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COLORED CHARTS

MECHANICS OF HEATING AND VENTILATING

WITH CHARTS FOR CALCULATION
AND EXAMPLES

BY

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PREFACE

The theories of heating and ventilating may be brought under three general headings:

1. The movement of fluids.
2. The transmission of heat or thermal functions.
3. The requirements of hygiene.

The first named would treat of the flow of the different heat carriers, and of air for ventilating, with special reference to the conditions met in modern practice. It would represent the mechanical side, while the heat transmission in its various steps, through building structure, as well as in generating or absorbing, radiating, and emitting heat, are more purely physical functions. In a work on the whole subject of heating and ventilating these two chapters could not be strictly separated, and the third one, the sanitary side, should also be considered in relation to the others. However, the present treatise is not intended to go beyond the first of these three equally important phases. Nor is it claimed to be an original contribution. It is presented as a collection of data from various sources, sifted, compared and selected with the idea of meeting certain needs of the day, and put into shape for ready application. At the same time, it has been attempted to give a comprehensive and explanatory outline of the subject, which may contribute toward friendlier relations between theory and practice, for the benefit of those who care, and make the data presented more useful to the student and engineer, and through them to the public.

The author is indebted to Mr. Fritz Schübeler, with Sulzer Bros., a pupil and former assistant to Dr. A. Stodola for a friendly discussion of some theoretical points, and to his office assistant Mr. Alfred J. Offner for the patient and faithful execution of the charts. K. M.

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MECHANICS OF HEATING AND VENTILATING

CHAPTER I

INTRODUCTION

The Need of Systematic Calculation.—It may seem to be self-evident that an intimate knowledge of the natural laws governing the movement of water, steam and air in conduits is indispensable to the heating and ventilating engineer. Nevertheless, it will be admitted, that the solution of the ever recurring problems of distribution and discharge is still attempted quite generally without a true and consistent application of these laws. The main obstacle to calculation on a physical basis lies probably in the lack of ready made data relating specifically to this field. Most of the information is obtainable, but more or less scattered, and not in convenient shape for practical use.

A superficial experience in the field may not convince that the scientific method will pay in engineering work of this character, but those among the profession who are in the habit of studying cause and effect will generally concede that a good share of the petty annoyances, of discomfort, and even of suffering brought about by indifferently designed heating apparatus can be traced to faults in the distribution of the heat carrier. In a ventilating system it is also the uncertain delivery, or what may be called the lack of control over air currents, that is responsible for much vexation, inefficiency and disappointment. Whatever may be the share of troubles that can be laid down to other causes, the ability to foretell the carrying capacity of a pipe line and to assure in advance a fair distribution of the fluid through a system of conduits of any complexity and length, would seem to be at least one of the prime elements that make for successful performance. But aside from the question of obtaining the results, a straightforward solution of the problem on a sound basis will obviate the need of excessive allowances to cover the lack of knowledge, and lead to economy in first cost. It will also con-

tribute to the efficiency of a plant in operation by avoiding waste through overheating and in other directions.

Thumb rules in general give only an average, which may not apply to the case in hand. They should not be relied upon for design and construction, but may serve for rough preliminary estimates. Neither will tabulated figures cover the requirements, if based on a fixed set of conditions, such as the pressure loss for a certain length, which hardly ever coincides with a particular instance. Such tables ought to be used only for approximate sizing.

For methodical calculation of a conduit it is essential to be aware that the fluid to be moved, or kept under control, must follow the laws of nature. All calculations relating to the flow should therefore be based, first of all, on formula giving adequate expression to the forces coming into play. This means that each case should be studied, not only with a view to applying the proper formula, but also in order to size up the various factors on which to base the computation. Theory and practice will be found to agree if this is done. Whenever they seemingly do not, some element has been left out of consideration or a factor has been misjudged.

It is most desirable also to have a clear grasp of the problem as a whole in the more complex cases of distribution. For instance, it is well to remember that the total pressure differences which we depend upon to create the movement are usually equal for all points of delivery, while the runs of conduit to these points are apt to be widely divergent in length and obstructing features. Diameters proportionate to quantity or to pressure drop for a uniform length would therefore present very unequal resistances for the intended flow. Since the total resistance inevitably conforms to the pressure available, the quantity delivered must depend on the relation of the resistance head to length of run. Technically speaking, the lines of *least resistance,* which the fluid is bound to take, must be made lines of *equal resistance* by giving them a different "hydraulic slope," or ratio of head to length, for each section of main, or branch, according to distances and other features. Translated into popular terms, long conduits must be eased by a low rate of resistance and short ones throttled by a high rate, in order to make up the same total.

To equalize the head for given volumes involves repeated calculation of each portion of conduit by formula on hand of

certain factors, to be assumed tentatively. The formulæ are mostly inconvenient, and their application takes more time and patience, even with the aid of complete tables, than is generally available. To make the process more popular, it must be simplified without material sacrifice in accuracy. The diagrams presented have been worked out with this end in view. They have been tried for several years in actual practice and the results have uniformly justified the labor of scientific calculation. According to the exigencies of the case, the procedure can be abbreviated, or carried further. Every engineer may evolve his own methods in using the charts, to suit his needs, but for successful and profitable use of the information presented it is necessary to be familiar with certain principles of hydraulics.

General Principles.—The movement of a fluid through a line of conduits always represents a certain amount of work performed, or of energy converted, which can be expressed in terms of heat or of mechanical power. If the drop in tension is known, the work can be computed in foot-pounds, as the product of volume of the fluid per time unit and pressure against which it moves, that is, by the product of distance and weight. Usually the volume is the given quantity from which the pressure is to be calculated. The latter is made up by the various resistances presented by conduits, and by the momentum of the flow, or the motion itself. The idea is best recalled by Fig. 1, familiar through all text-books on hydraulics. It may serve here for reference and explanation of the terms and symbols used in this treatise.

According to the law of gravitation it is always the height H through which a fluid drops, or a pressure P equivalent to the weight of that column for a density w, that determines V, the theoretical velocity of fall or of frictionless discharge through a rounded orifice. The actual velocity in a conduit is always smaller, owing to the friction and the resistances offered by throats, elbows and other features, which take a certain definite head to overcome them, and may be expressed in terms of pressure or height p_f and p_r, or h_f and h_r. The difference between H, the total head available, and $H_f + H_r$, the combined resistances presented by the conduit, gives H_v, the net effective head exerted by the moving fluid, or the velocity head, as measured by the Pitot tube. From this can be figured the actual velocity of discharge $v = \sqrt{2gH_v}$, which gives the volume of the fluid Q for the conduit area a.

The resistance by friction and obstruction or local resistances, both representing lost motion, retard the flow to a greatly varying extent, according to the length and character of the conduits. Either of these items may be the major portion of the total head and control the situation. In other words, the final speed of delivery for the same head H and volume Q may range from the

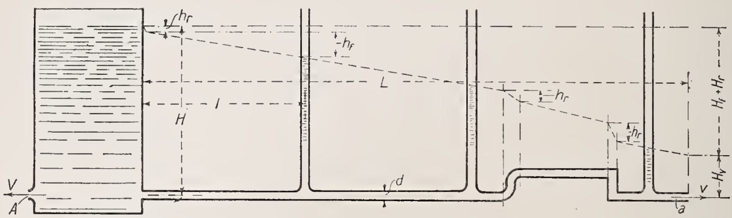


FIG. 1.—Diagram of the flow of liquids in conduits and explanation of terms and symbols.

H = Total or theoretical head, in ft.
H_f = Total head to overcome friction, in ft., or Σh_f.
H_r = Total head to overcome resistance, in ft., or Σh_r.
H_v = Total head to create actual velocity v, in ft per sec.
w = Density of fluid, or weight per cu. ft.
P = Hw = Pressure in lb. per sq. ft.

V = Theoretical velocity in ft. per sec. $= \sqrt{2gH} = \sqrt{2g\frac{P}{w}}$

v = Actual velocity in ft. per sec. $= \sqrt{2gH_v} = \sqrt{2g\frac{P_v}{w}}$

Q = Volume of fluid per sec. $\frac{Q}{v} = a \quad \frac{Q}{V} = A$

W = Weight of fluid per hour = 3600 w Q.

d = Diameter of conduit.
c = Circumference of conduit.
a = Area of conduit in sq. ft. $= \frac{Q}{v}$
A = Equivalent area in sq. ft. $= \frac{Q}{V}$

l = Length of conduit in ft.
f = Coefficient of friction.
r = Factor of resistance.
g = Acceleration (32.16).

full theoretical velocity V down to nothing, while the conduit increases from the frictionless orifice for V, or the equivalent area A upward, indefinitely, depending on the resistances opposed to the flow.

The basic formula for friction applies equally to any fluid. The loss of head for a run of conduit of uniform cross-section, according to the generally accepted theory, is expressed by

$$h_f = f\frac{v^2}{2g}\ \frac{lc}{a}$$

for any shape or cross-section of conduit, or

$$h_f = f\frac{v^2}{2g}4\frac{l}{d}$$

for round pipes only.

The velocity corresponding to the area and volume transmitted, $v = \frac{Q}{a}$, is assumed to be the mean velocity within the conduit, the flow being more rapid in the center than at the circumference or surface of contact and always more or less turbulent above a certain speed. The friction naturally depends on this turbulency, or the internal losses of motion, which are known to vary with different ratios for $\frac{a}{c}$, the hydraulic radius. It is also influenced materially by the character of the contact surface, the viscosity, and the velocity itself. All these factors have been shown to bear on the coefficient of friction f, under conditions met in ordinary practice, but to a varying extent for water, steam and air. The above formula is therefore only applicable in connection with additional data giving the proper value of f for the fluids in question and for the kinds of conduit used. The values given for this coefficient by investigators for various conditions is more or less conflicting, and much of the data does not apply to speeds and conduits met in heating and ventilating practice, hence the engineer is often left in doubt as to the proper value to be used in a given case. Other formulæ which express these variations directly by composite constants are inconvenient. This difficulty is avoided by Tutton's general form

$$v = CR^xS^y$$

which includes the bearing on f of the hydraulic radius R, and the slope $\frac{h}{l}$ or S, through the odd exponents x and y, and therefore can be made to apply at least for one kind of contact surface and the same viscosity, that is, it will be correct for a range of areas and velocities covering an entire field, either water, steam or air. If we substitute $\frac{a}{c}$ or $\frac{d}{4}$ for R, $\frac{h}{l}$ for S and call the constant $\frac{1}{C^{\frac{1}{y}}} = f$, the formula reads

$$v = f\left(\frac{d}{4}\right)^x \left(\frac{h}{l}\right)^y \text{ or } h_f = f\frac{v^{\frac{1}{y}}l}{d^{\frac{x}{y}}}$$

This formula is, of course, not convenient for working purposes, but the values derived from it can be charted in straight lines on logarithmic paper. The exponents of hydraulic radius and slope are thereby expressed by the slant of the lines of diameter and velocity, which differs, as stated, with the viscosity and roughness of surface. A glance at any of the charts will show at once the interrelation of pressure, volume, velocity, and area. Two of the factors known will determine a point of intersection and give the value of the other two. Thus, for instance on Chart IX, a volume of 25 cu. ft. of air per second moving at 12 ft. velocity will require an area of 300 sq. in. and cause a pressure loss of .27 lb. per sq. ft. per 100 ft. of length.

Separate charts are necessary to cover the flow of water, steam and air, but this would be desirable in any event, in order to give the quantities in the customary measures. The three principal fields have been further divided with the idea of presenting distinctive classes of work to best advantage, each chart being worked out only for the needed range at the largest scale practicable for publication. It has been possible in this way to bring much data together, to assign to it its proper sphere, and to make it ready for use. The sources of information on which the charts are based are stated in the chapters on water, steam, and air.

Reliable and pertinent information on local resistances due to various forms of obstruction is much less complete than that for friction, although it is often quite as important for calculation, and should receive equal consideration. Usually, an item of local resistance is expressed as a function of the velocity head

$$h_r = r\frac{v^2}{2g}$$

wherein r is the coefficient of resistance intended to measure the degree of obstruction presented to the flow by various features, such as bends, irrespective of size. According to experimental evidence, this expression is not strictly correct, and should be considered only as a general approximation. Present data on the point is too scant to permit definite conclusions, but there are sufficient figures available to indicate that the loss of head also depends on the diameter of conduit and the velocity. Considered as a degree of friction, which is also loss of motion through uneven conduit surface, it seems logical that such resist-

ances would follow the same general law and might be expressed in a way similar to Tutton's formula. This theory is strengthened by the fact that the coefficients of contraction are known to vary with diameter and velocity. All forms of obstruction involve more or less contraction of the stream, hence it is reasonable to assume that the loss of head may be stated by a formula similar to Tutton's

$$h_r = C.r \frac{v^{\frac{1}{y}}}{d^{\frac{x}{y}}}$$

with the exponents for v and d to be determined for different classes of work. This mode of expression would require separate charts giving the resistance head for any diameter and velocity. Inasmuch as the bearing of v and d on the factor appears to be less pronounced than it is for friction, a fair approximation can be obtained if the formula is simplified to read

$$h_r = C.r \frac{v^{\frac{1}{y}}}{2g}$$

and by taking the exponents for v equal to that for friction. This form permits the losses of head by local resistances to be plotted on the same chart, by lines of different colors or character, parallel to those for diameters, and giving the values for individual items, not for any area, but for any velocity. The readings are thus made decidedly more convenient, as the pressures for friction and several features of obstruction for a run of even area may be read from the same velocity line, without straining of the facts beyond the probable limit of error. These items of resistance, at best, must remain an uncertain quantity owing to the individualities of commercial forms and features. As presented, the results should be, on the average, at least as accurate as general data applied to special cases.

In order that r should express the relation of the resistance to the corresponding velocity head, the constant C has been made to equal $\frac{v^2}{v^{\frac{1}{y}}}$ for a velocity at which the generally accepted coefficients of resistance would fairly apply. For $v = 1$, C would be $\frac{1^2}{1^{\frac{1}{y}}} = 1$ and the head $h_r = r\frac{v^2}{2g}$, but the values for all higher

velocities would be less than the function of $\frac{V^2}{2g}$. It is probable that the average of experimental data on r has been derived at higher velocities and is more nearly correct at 10 ft. p. s. On this assumption the constant C has been made to equal about $\frac{10^2}{10^{\frac{1}{y}}}$ in the applications of the formula to water, steam and air, thus giving higher average values throughout, which are expressed in the formula for h_r by different constants.

The velocity head is charted on each diagram by a line of definite slope, giving at a glance the accurate value of $H_v = \frac{V^2}{2g}$ or of $P_v = w\frac{V^2}{2g}$, for any speed. *Vice versa*, from a known rise in a Pitot tube we can read directly the velocity at which the fluid must be moving. The line is also most convenient for the frequent readings of the dynamic head, or momentum of the flow incidental to duct calculation, especially in connection with its bearing on the flow at junctions or through contractions and enlargements of conduit, and wherever the correlation of static and dynamic head may enter into the problem. It can also be used conveniently to determine the so-called equivalent area A corresponding to the pressure at any point of the conduit. The intersection of this line with that of H or P gives the theoretical velocity V, and the area corresponding to that speed and the volume is $\frac{Q}{V} = A$, which must be distinguished from $\frac{Q}{v} = a$. The former, A, represents the smallest, or theoretical area through which the volumes could pass, while the latter, a, designates the actual area, which must be greater, according to the resistance to be overcome. This equivalent area or orifice is commonly assumed to be an opening in a thin plate, with sharp edges, giving a contracted stream and allowing for this contraction according to the coefficient of efflux. Blaess,[1] who applies the term to his method of duct calculation, suggests the rounded frictionless outlet as a more definite basis. In the present treatise A refers also to the theoretical frictionless orifice.

The total head is represented on the charts in different ways, according to the field of application. With forced calculation of

[1] Dr. Ing. Victor Bläss, "Die Strömung in Röhren," Oldenbourg.

air and water it gives the power requirements, and, incidentally, the theoretical head that a machine must produce, together with the corresponding velocity and blast or equivalent area. That is, it provides the correct basis for the calculation of centrifugal or other machinery for propelling the fluid. On the table for gravity hot air heating and vents, the available total head, varying in each case, is charted for a liberal range of conditions. For steam and hot water heating where the total head is usually predetermined, an auxiliary table is worked out for convenient approximation, or for tentative sizing, as a basis for final accurate calculation.

The charts, in addition, give the special data for balancing gravity hot water heating systems, and the principal factors for correction applying to unusual and variable conditions, as far as it will pay to consider them. They give also the means for rapid, but consistent, approximation, and for the quick reading of the simple relations of area, speed and quantity.

The true usefulness of the charts depends, of course, upon a clear grasp of the problem in hand, as for instance the delivery of fixed volumes at certain points with a uniform pressure drop, which would be a problem in equalization, or of delivery against different heads, a case of balancing. The charts will be found to provide the necessary facilities to apply the mechanics in every-day practice.

Previous Use of Logarithmic Charts.—Similar charts have been worked out for various special purposes more or less related to heating and ventilating, and covering small parts of the field. Among these may be mentioned the diagrams on the flow of water in channels by Prof. I. P. Church[1] and the diagrams by Schoder[2] and Bleich[3] for the flow of water in pipes. Birlo, a German heating engineer years ago applied the principle to hot water heating. These tables are now reissued in a new form, though not on a logarithmic scale, by Schweer.[4] Gremmels has

[1] "Diagrams of Mean Velocity of Uniform Motion of Water in Open Channels." Irving P. Church, C. E. John Wiley & Son.

[2] "A Diagram for the Flow of Water in Pipes." E. W. Schoder, *Engineering Record*, Sept. 3, 1904.

[3] "Hydraulic Diagrams for Circular Pipes." Samuel D. Bleich, *Engineering Record*, Vol. LVI, No. 22.

[4] Schweer, "Graphische Rohrbestimmungs Methode für Warmwasserheizungeen."

applied the principle to low pressure steam heat.[1] Gramberg has compiled a series of tables for hot air and water heating which include also data for giving the total head available in gravity work.[2] A collection of similar tables for water pipes and channels has been issued by A. Gremand.[3]

The charting on logarithmic paper for such purposes is capable of being improved and extended to other lines of work. In fact, the possibilities of this method of calculation are only beginning to be appreciated. The idea underlying the slide rule is here applied to specific purposes, permitting the direct reading of values, otherwise calculated by formula, for a given range of factors which can be extended indefinitely by mere elongation of the lines. No auxiliaries, such as scales, planimeters, dividers or other tables, are necessary. On the whole, the graphic method should contribute materially to efficiency in engineering by encouraging the true and yet practicable solution of a problem.

The intention has been to make these diagrams cover as closely as possible the special requirements of engineers for heating, ventilating and similar work and with particular reference to modern building practice. By way of explaining the application of the charts to best advantage, a general treatment of the mechanics relative to this special field is presented in the following pages.

[1] "Tafel zur Berechnung der Druckverluste in den Dampfleitungen für Niederdruck Dampfheizungen" Gesundheits Ingenieur, 1905.

[2] See "Gesundheits Ingenieur," No. 27, 1907.

[3] "Graphische Tafeln zur Bestimmung der Dimensionen von Wassereitungen und Kanälen."

THE FLOW OF WATER

CHAPTER II

THEORY OF THE FLOW

Properties of Water.—The weight and volume of water at different temperatures, the specific heat and the boiling-points under various pressures, are presented by a diagram in place of the customary table of properties, which will permit readings down to any fraction desired in practical work and without the necessity of interpolation.

The density curve is based on Nystrom's computations from experiments by Kopp, as given in Supplee's "Mechanical Engineering." The boiling points are given for various pressures, expressed in water columns at respective temperatures and in air columns or elevation above sea level. The equivalent pressures in pounds per square inch and vacuum in inches can be found on the steam table. Separate scales are given for pressure per 1 ft. of height, and reciprocals, from which the accurate pressure for any height, at any temperature can be figured. The lines for total heat above 32° F. are made to correspond with the recent investigations by Diederici,[1] which follow Regnault's figures closely at least up to about 300° F. The unit of heat is taken to be the mean value between the freezing- and boiling-points. The B. t. u. therefore give the accurate amount of heat for any rise in temperature by subtraction.

Friction in Pipes.—Of the numerous formulæ for the flow of water in iron pipes, Schoder's application of Tutton's general expression, quoted in the introductory chapter, consistently renders the bearing on the friction head of the pipe diameter and the mean velocity of the flow. For wrought iron pipe he finds[2]

$$v = 174R^{.54}, \text{ or } h_f = .00038\, l\, \frac{v^{1.86}}{d^{1.25}}$$

The exponents for v and d are the result of an extended series of

[1] "Die Calorischen Eigenschaften des Wassers und seines Dampfes." *Zeitschrift des Vereins Deutscher Ingenieure*, 1905, p. 362.

[2] See *Engineering Record*, Sept. 3, 1904 and "Trans. A. S. C. E.," Vol. LI, p. 308.

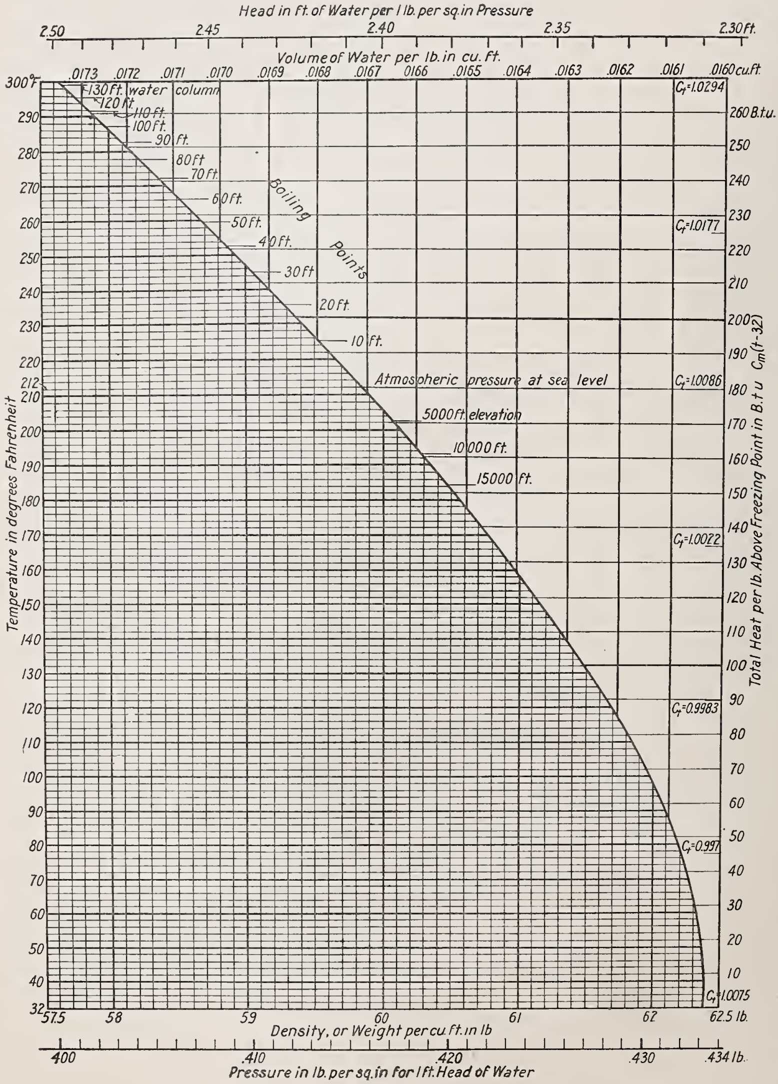


DIAGRAM A.—Properties of water

experiments, and may be accepted as covering the variations of the coefficient of friction given by various authorities for the familiar formula $h_f = f \frac{v^2}{2g} \frac{l}{d}$. Schoder's value for C is therefore a constant for one kind of pipe, and his exponents for R and S are correct for the same viscosity. When modified to read

$$h_f = .0245 \frac{v^{1\cdot 86}}{2g} \frac{l}{d^{1\cdot 25}}$$

Schoder's form will show its relation to that used by Weisbach. It will be apparent that for v = 1 ft. and d = 1 ft., the constant .0245 should be equal to the old variable coefficient. Weisbach puts f at .0316 for 1 ft. velocity at any diameter, but his experiments were made on small sizes (3/8 in. to 1 in. only),[1] for which the newer formula will give even somewhat higher values.

To provide a safe margin for commercial pipe, as used in heating work, and assumed to be in fair condition, also to allow for the extra resistance of ordinary couplings, the constant .00038 has been rounded up to .0004, or to $\frac{.0257}{2g}$. The friction losses on the charts are plotted on this basis with the velocities figured for the actual inside diameter of standard weight wrought iron or steel pipes.

For ordinary ranges the friction heads will be found to check fairly well with those derived from Lang's, Weston's, D'Arcy's, Fanning's, Tutton's, Hazen-Williams' and Pelton's rules and tables, but due allowance must be made in comparing, for the varying range and conditions for which the figures are intended to apply. The charts also agree closely with German data for heating practice, except for small pipe sizes, for which Rietschel in his treatise gives a somewhat lower friction head. Later experimental data from the same authority show the discrepancy to be smaller, and the most recent information gives lower values throughout, but presumably for new pipe, tested under ideal conditions. The formula by Ganguillet and Kutter, which is widely recognized, on the other hand, gives still higher friction head for the same diameters, probably allowing for a greater amount of fouling than is met with in heating practice. Unwin's coefficients, as published in his recent treatise on hydraulics,

[1] See "Weisbach's Mechanics," p. 867.

give smaller friction head throughout, based, however, on new wrought iron pipe.

Considering the number of factors that come into play, a closer approximation of the data available is hardly to be expected. Even within the limits of heating practice the roughness of contact surface will change, and modify the constant. It is probable, also, that the velocity, especially in gravity hot water circulation, is often within or below the critical stage at which the motion of the water is supposed to be parallel, and the friction head proportional with the speed. The range of velocities at which this might occur depends, however, on pipe diameter as well as the frequency of fittings and joints. The latter will practically always keep the water in turbulent motion, on which formulæ on friction head are based. Hence it is not advisable to base calculations upon such favorable but uncertain phenomena.

The changes of viscosity with the temperature of the water, according to recent investigations by Biel[1] would seem to have an appreciable effect on the friction head, also depending on diameter and velocity. For heating work this would have greater bearing than is the case in other fields. Unfortunately, experimental data on this point is very incomplete. Saph and Schoder have demonstrated the fact for the range between 40° and 72° F. Rateau,[2] on the other hand, has experimented at high temperatures, and pressures near the point of evaporation, at which he finds the sudden variations of density, due to pressure changes to make the losses of head "extremely interesting." They would be difficult to determine for practical purposes even if more data were available. Except for artificial circulation, this range is outside of conditions met in hot water heating, but the experiments in general point to the fact that the friction head decreases about evenly between the freezing- and boiling-points.

Whatever the weight of these considerations may be, it will not pay to express them by additional lines or corrections. The charts will be more serviceable in the simplest form, intended to represent average conditions and to be within the probable limit of errors from other factors. In problems of equalizing or distribution such differences would affect all parts alike, and

[1] "Mittheilungen über Forschungsarbeiten," Heft 44, Verein deutcher Ingenieure.

[2] A. Rateau. "Recherches experimentales sur l'ecoulement de la vapeur d'eau, suivies d'une note sur l'ecoulement d'eau chaude."

would have no bearing on the result. In problems of discharge, a small error in the friction, as one item in the total head, will not mean the same, but a still smaller discrepancy in the volume carried.

Local Resistances.—The assumption explained in the general introduction, that the coefficient of resistance decreases with the velocity, as the coefficient of friction, when applied to water, make the formula for the loss of head $h_r = 1.38\ r\frac{v^{1.86}}{2g}$. The constant 1.38 represents the ratio of $\frac{v^2}{v^{1.86}}$ for $v = 10$ ft., so that h_r would be a function of $\frac{v^2}{2g}$ at that velocity, and would assume somewhat higher values for lower speeds, and *vice versa*. Taking for instance on Chart I the resistance for an open bend, for which $r = .42$, we find the loss of head at 10 ft. to be .65 ft. which equals $r\frac{v^2}{2g}$. At 5 ft. the chart gives $\frac{v^2}{2g} = .38$ ft., and $r\frac{v^2}{2g}$ would be $.42 \times .38 = .16$ ft., while the direct reading of h_r for that form on the line of 5 ft. velocity is .175 ft. As stated previously, this increase in values is justified, partly on experimental grounds, and also because the factor r should be somewhat greater for small size fittings in which the velocities are generally lower. The charts are thereby simplified while presenting values at least as accurate, on the average, as usually obtained by formula with coefficients to be selected in each case. The practice of expressing resistances of elbows or other features in equivalents of pipe lengths, is good for only one size, and when expressed in diameters it is strictly correct only at one velocity. These rules; therefore, are likely to be less accurate than the values obtained from the charts.

Little data is available giving the coefficients for the standard designs, or even the average types of commercial fittings and valves. Weisbach's formula[1] for bends and deflections of different radius, at different angles and his coefficients of contraction, apply to the ideal shapes rather than to commercial forms. His own findings indicate that the common style of fittings, as shown on Fig. 2, with shoulders exposing the butt end of pipes, and presenting a considerable and sudden enlargement of area,

[1] See Weisbach's "Theoretical Mechanics" p. 896.

involves materially greater resistance than the ideal shape would offer.

The factors of resistance for the various forms of obstruction, as given on the diagram, has been obtained by adding to Weisbach's theoretical figures the probable effect of such retarding features. For recessed and for flanged fittings, flush with the piping on the

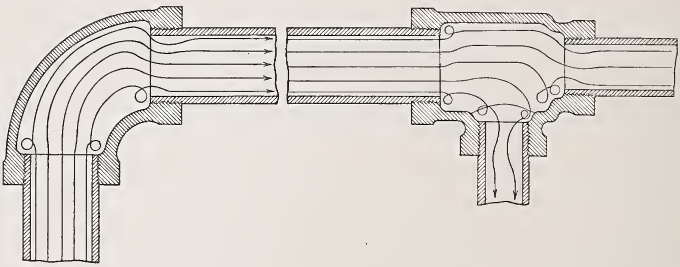


Fig. 2.—Typical cast iron fittings.

inside, this does not apply. In such cases about 25 per cent. may be deducted from the head as charted.

The values for the factor r thus obtained check fairly well with such meager data as can be found for typical shapes and designs, but are higher throughout than those adopted by German authorities which seem to be based on Weisbach's experiments. Accord-

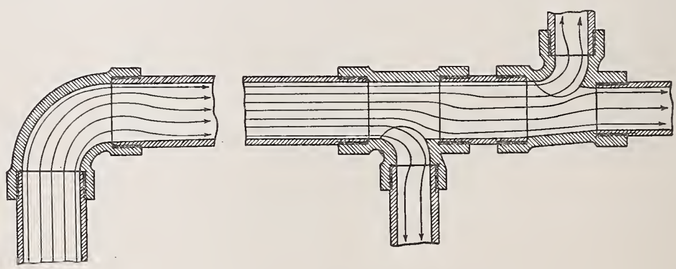


Fig. 3.—Recessed sweep fittings.

ing to the latest information on experiments in progress these German factors appear indeed to be taken too low.

Inasmuch as they can only be approximate, the various factors have been rounded up, reduced in number and grouped together in order to avoid multiplicity at the expense of clearness. Thus, it happens that a T fitting without reduction on the run is represented to give the same resistance, straight ahead, as the long

sweep elbow. An ordinary T fitting reduced by one size on the run, will be about equivalent in obstruction to an open bend, and a T reduced two sizes, to a close elbow. For greater reduction r may be assumed to equal about the head lost by the partial expansion and subsequent sudden contraction of the area within the fitting which approaches the velocity head in the smaller end. If the reducer is a taper piece, the contraction is partially avoided, and the same factor as for a straight tee will apply. Likewise, for a reducing elbow the resistance is practically that for an ordinary elbow, based for safety on the velocity at the small end. A junction at right angles includes the velocity head which must be created again out of the static head, in ordinary fittings without gradual change of direction. This item is to be added to those for the branch, as indicated by the line of resistance on the diagram, which is always drawn through the part in question. The factor for radiators is a composite, including the velocity head to be entered, in addition to the estimated obstruction of two angle valves and the sections, all of which generally occur together in the calculation. If one valve and one elbow are used, the resistance can be figured by subtraction, but this will hardly pay since the difference is a small fraction of the item as a whole. With uneven sizes of flow and return pipes, it is convenient and accurate enough to take one-half of the reading for the velocity of inlet, and one-half for that of the outlet, giving together the mean value of the two. The velocity head is also included in the factor for boilers, which are assumed to present somewhat more obstruction than a simple tank, allowing for the friction in nipples or headers. Since the loss of head is based upon the outlet velocity it is advisable to round up the figure for small boilers with large outlets, giving a proportionately high velocity through the sections. Coil heaters with headers, having the water passage arranged in series of liberal area, will present about the same resistance as a radiator. When made up as a continuous single pipe coil, the latter it to be figured as part of the pipe system. Heaters with coils for steam may be assumed to present the same resistance as a tank. In every case the factor r must be understood to refer to the velocity at the outlet, as indicated by the dash lines, drawn through the respective parts.

The factors for other forms of obstruction can readily be estimated by comparison with those at hand. Elbows with 45

degrees deviation give somewhat more than half the resistance of a 90 degree turn of same radius, and larger angles less, when in the same plane of deflection. For ordinary practice it is permissible, however, to assume the factor for an ordinary, close 45 degree elbow to be equal to that for a 90 degree medium radius elbow.

Velocity Head.—As an item in the sum of losses making up the total head, the velocity head is generally understood to be the balance expended in discharging the fluid. In heating practice the water is nearly always returned, or moving within a closed circuit, for which this balance remains unexpended while the movement takes place. The velocity head in the ordinary sense, as the resultant force of movement, does not enter, therefore, into the calculation of the total, except where the flow along the circuit is interrupted by enlargements of area, as represented, for instance, by a boiler or radiators in which the movement practically stops. The losses of head incidental to such features must be considered. As mentioned above they are included in the factors charted.

For unusual conditions, such as interruption of the movement for parts or the whole of the circuit, the momentum of flow to be created will temporarily bear on the result, but in ordinary practice it is not necessary to consider this.

The dynamic head and its relation to the static head should nevertheless be borne in mind. It should be calculated sometimes for the purpose of estimating the factors of resistance for odd forms of obstruction, and considered also in its bearing on the flow at junctions with very unequal velocities.

In the case of sudden enlargement of pipe area the factor is expressed by the Carnot-Borda formula, which makes the loss $h_r = \frac{(v - v_1)^2}{2g}$. If the enlargement is gradual, the loss may be reduced to a negligible item. The same applies to contraction of pipe area, for which the resistance can be avoided by rounding the throat, the idea being to convert static into dynamic head, and *vice versa*. These factors, like the resistance in a nozzle or a "Venturi" tube, may differ considerably with the direction of the flow for the same shape. Wherever possible, the shapes should conform to the natural flow. In other words, they should be designed to avoid eddies or loss of motion, and thereby practically eliminate resistance.

The velocity head maintaining at junctions will modify occa-

sionally the losses for branch T's at right angles, as taken into account by the charted item of resistance. The latter gives an average value of r = 1.33, allowing for contraction as well as loss of motion. This factor will apply when the flow in the branch is not materially slower or faster than that in the main, that is, when the component of velocities does not deviate radically from the direction of the branch piece. With higher velocity in the branch, for which the natural direction of outflow nears the right angle, the factor may be taken smaller, since the motion lost in the main is relatively smaller, and that in entering the branch is fully taken into account. If, however, the flow in the branch must be at decidedly lower velocity, a right angle fitting will produce a relatively greater resistance, as the natural outflow is tending in a forward direction which the ordinary T fitting will not meet. It is proper in such cases to allow for greater loss of head, or to use sweep or Y fittings to ease resistance. The factor

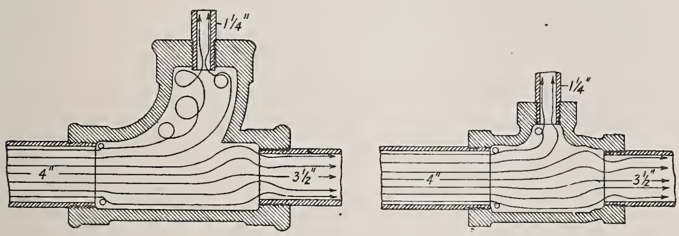


Fig. 4.—Two types of branch T's.

is thereby reduced and may be assumed as equal to that for a turn of corresponding radius. This rule will not apply, of course, when the side outlets of such fittings are of the full size of the main, and bushed down or reduced suddenly by a blank head threaded for a smaller size. For such fittings, in fact, a greater allowance should be made than for ordinary T's, since the latter have slightly eased throats, while the blank head creates eddies in a dead space with full contraction at the outlet. This form of branch piece, improperly called a "fitting," is a very uncertain factor of resistance, since the attempt at deflection by a sweep, with subsequent contraction, is worse than useless. The opportunity for air to collect is a further drawback of such shapes.

Stated in a general way, the relation of static and dynamic head at junctions should be considered, and at least roughly expressed by the style of fitting. Wherever the factors for

average conditions would not fairly apply, a pattern should be selected that will suit the case. With high velocities near the end of a line, and little static pressure left it is advantageous, for instance, to utilize the *vis-viva* or force of the main flow as much as possible for the branches by styles of fittings easing the resistances. This is recommended to reduce the uncertainties of distribution, quite aside from the waste of energy that may be involved in certain cases. On the other hand, when the static head is comparatively great, it is a mistake to try to induce distribution by deflecting the currents rather than by using proper area of pipe. The former mode is often wrongly attempted in gravity hot water heating. On mains where the velocities are

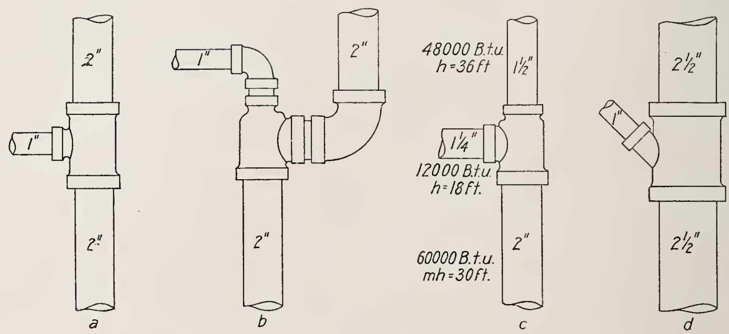


Fig. 5.—Methods of running branches from risers.

greatest and the losses should be kept down, it will pay to ease the flow by sweep bends, by the use of gate valves and other means. On rising lines on the other hand, where the head available for the upper stories is usually ample, it is proper to use such pipe sizes that will make up the desired, easily figured resistance, rather than obstructing the flow by extra turns as indicated by sketch b on Fig. 5, or by freak fittings of unknown effect.

With the arrangement "a" the 2-in. pipe leading upward would be grossly disproportionate for the requirement, when we consider the greater head available, and the decreased velocity. Fitted up in style "b," the extra turns on the main riser still give too much capacity for the upper floors, creating a greater velocity than necessary for conveying the heat, and thus putting the branch under a disadvantage, unless throttling devices are relied upon. Form "c," with the sizes proportioned with due consideration for the heads available at the main and the branch, will

give the required capacity without adjustment, with less fittings, less pipe, less heat loss and less expense for labor. In cases where the main cannot be reduced and the resistance in the branch must be kept down, Y fittings as shown by the arrangement "d" are indicated. With an ordinary T as shown in "a" the head available for such a branch would be further reduced and would become uncertain, especially when handicapped still more by excessive flow to the stories above.

Total Head.—The sum of losses in head by friction, resistance and motion, $H_f + H_r + H_v = H$ enters in nearly every case into the calculation of conduits.

For a simple pipe line with free discharge on one end, the total head gives the pressure or height of water column necessary for the desired delivery, or if the total head H is the known quantity, the difference between it and the combined friction and resistance heads $H_v = H - (H_f + H_r)$ will give the velocity head or net pressure available for discharge. The chart gives immediately the corresponding velocity and volume of water for the pipe size on which the losses of head $(H_f + H_r)$ are based.

For an open distributing system, when H is uniform for all points of delivery, it must be determined as a basis for calculating the several branches, which should be equalized, or propor tioned so as to make up the same total loss when discharging the desired amounts.

In the case of gravity circulating H is given by the relative levels of heating and cooling surfaces, and the differences of temperature to be maintained. This available head is to be figured for each circuit, so that the sum of resistances can be made to balance it by selection of the proper pipe sizes and forms of local obstruction.

With forced circulation, usually a closed distributing system, the total head is sought for equalizing distribution as well as for determining the proportions of the pump for propelling the water, and for estimating the motive power. The loss of head by friction, eddies, internal leakage around the rim, or slip, and changes of velocity within the pump, are in reality a part of the total to be overcome. It would be impracticable to express these hydraulic losses in the pump as a function of the head to be produced by the wheel, since the latter depends upon the nearness with which the theoretical requirements, that is, the correct relation of dynamic and static heads, can be approached by the

commercial article to be used. The pump must produce a total head including these internal losses. Its speed and efficiency under given conditions should be known by the manufacturer, and it is proper, therefore, to call for a certain volume to be moved against a certain head, which head should always be understood to include the sum of losses from the outlet of the casing to the point of delivery, plus that of the suction or the return pipe up to the inlet. In order to avoid undue losses through discrepancies between the sizes of mains and those of the ports, it is best to state the size of these mains, or the velocity at which the water is to enter and leave the pump, thereby defining also the dynamic head and its relation to the static head.

For the common type of single stage centrifugal pump the peripheral speed will average about 1.15 V, or 15 per cent. in excess of the theoretical velocity corresponding to the total head outside the pump. The velocity at which the water should leave the wheel and the pump depends upon the relation of static and dynamic head desired, which is conveniently established by the charts. High static head or back-pressure against the flow and low velocity involve large area of outlet and call for a wheel with blades at a small angle to the tangent, imparting lower speed at higher pressure. High velocity and low back pressure would favor radial blades for which the discharge velocity at the rim may be nearly equal or even greater than the tip speed of the wheel. To what extent this velocity should be reduced in the casing or converted into pressure without excessive loss by cones at outlet and inlet connections to piping, would depend on commercial considerations.

These general conditions should be recognized in order to assure a fair hydraulic efficiency for a circulating system, but the exact relations of theoretical velocity, tip speed and discharge velocity at rim and ports must naturally be governed by the commercial sizes available. As stated, the selection should finally be made by the manufacturer.

Motive Power.—The power required to propel a given amount of water per hour through a system of conduits is equivalent to the work of lifting its weight up to a height balancing the total resistance to the flow, which the chart gives in feet of water column. In foot-pounds per second this will be

$$\frac{WH}{3600}, \text{ hence in h. p.} = \frac{WH}{3600 \times 550}.$$

This energy represents the net output required of a pump. The motive power necessary to drive the same must include its mechanical losses, also the internal hydraulic losses, all to be overcome by greater power delivered to the shaft. The losses of motion and leakage are to be made up by larger diameter, 1.1 to 1.25 times the theoretical. This item alone will often increase the power need from 1.1^2, up to 1.25^2 of the theoretical, or from 21 per cent. to 57 per cent. additional. The friction of wheel and casing, independently of the losses of motion, will require also from 33 per cent. to 67 per cent. extra power, according to perfection in design and execution. The mechanical losses are a small percentage. The power delivered to the shaft will therefore have to be from 60 per cent. to 135 per cent. greater, for pump efficiencies varying between 67 per cent. and 42 per cent. Allowing for the uncertainties of the motors themselves, it is proper to purchase the motive power on the basis of 2 to 2 1/2 times the value obtained by the chart, as stated. Only for large units, specially designed, or well suited to conditions, is it safe to take less than twice the theoretical h. p.

CHAPTER III

FORCED HOT WATER HEATING.

Chart I.—The chart for forced circulation is intended principally for district heating from a central station, and wherever heat is to be transmitted by hot water at long distances, for which the flow must be created by mechanical means.

When heat is conveyed by water, only a margin of the B. t. u. contained in the fluid is transmitted. Economy in operation will therefore demand a return circulation of the medium under a certain drop in temperature. This drop, as well as the mean temperature, depends upon the conditions under which the heat is absorbed and emitted, that is, upon the temperatures available for generating and the requirements as to intensity of the heat at points of delivery. Other things being equal, a higher flow temperature will permit transmission of the heat with smaller volumes, at a greater range, while lower flow temperature, with relatively warmer return, involves greater volumes. The drop in temperature, defining the quantity of water to be kept in motion, ranges between 10° F. and 30° F. in ordinary practice. The chart gives the heat transmitted per 1° difference. Thus, if 500,000 B. t. u. are to be transmitted with the water cooling off by 10° F., the weight of water circulated, will be 50,000 lb. per hour, or 16,667 lb. for 30° F. drop between flow and return. For a range of 20° F. the amount of heat transmitted can be read directly on the scale of gallons per minute, each gallon carrying in round numbers 10,000 B. t. u. per hour.

The friction is given for a unit of 10 lin. ft. of standard wrought iron pipe, and the resistances for the ordinary types of fittings and apparatus, the losses of head being expressed in feet of water column. The necessary data is added for correcting the volume to allow for heat losses, and for correcting the head when influenced by temperature.

The supplementary chart is intended for determining the theoretical velocity, area and power requirement, from the total head. A separate scale is given for conversion of the head in

V

V

H

T

H

E

T
A

C

A

p

a
m

t
w

CHART I

FORCED HOT WATER HEATING

THROUGH STANDARD WEIGHT IRON PIPES

Water circulated in lb. per hour $W = \dfrac{\text{B.t.u. per hour}}{t-t_1}$.

" " gal. per minute $= \dfrac{\text{B.t.u. per hour}}{8.05\times 60\times(t-t_1)}$ at 200° F.

" " cu.ft. per sec. $Q = \dfrac{W}{3600\times\dfrac{w+w_1}{2}} = .00000463W$ (200° F.)

Velocity of flow in cu.ft. per sec. $v = \dfrac{Q}{a}$.

Head in ft. to overcome friction $h_f = .0257\dfrac{v^{1.86}}{2g}\times\dfrac{l}{d^{1.25}}$.

" " " resistance of obstruction $h_r = 1.38r\dfrac{v^{1.86}}{2g}$.

" " create velocity $h_v = \dfrac{v^2}{2g}$.

Total head in ft. to be produced by pump $H = H_f + H_r$

" pressure in lb. per sq.in. to be produced by pump $P = \dfrac{w}{144}H = .42H$ (200° F.)

Horse-power to move water against total head $= \dfrac{W.H.}{3600\times 550} = .000000505\,W.H.$

Energy in B.t.u. p.h. to move water $= \dfrac{2545\,W.H.}{3600\times 550} = .00128\,W.H.$

Theoretical velocity giving total head of delivery $V = \sqrt{2gH}$.
Approximate peripheral velocity of pump wheel $= 1.1V$ to $1.25V$.

Orifice in sq.ft. corresponding to theoretical velocity $A = \dfrac{Q}{V}$.

Approximate area of pump inlet and outlet in sq.ft. $= 1.1A\sqrt{V}$ to $1.25A\sqrt{V}$.

For a drop of 20° F., between flow and return mains ($t - t_1 = 20$) one gal. per min., will transmit heat at the rate of about 10,000 B.t.u.

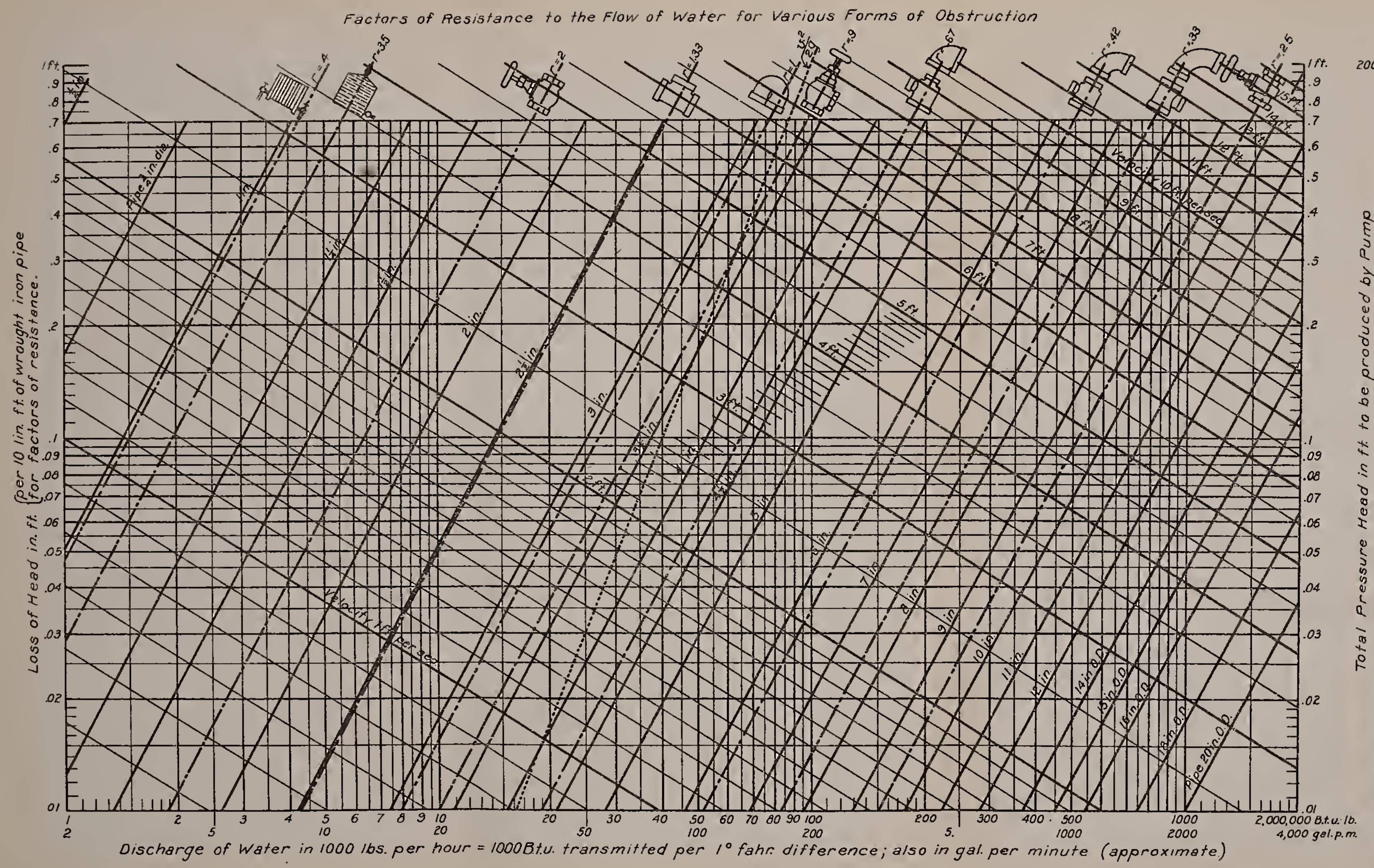
The B.t.u. transmitted should include the heat lost in transit as far as affecting the volumes for the run to be figured. To estimate this loss for a mean temperature of about 200° F., take roughly for

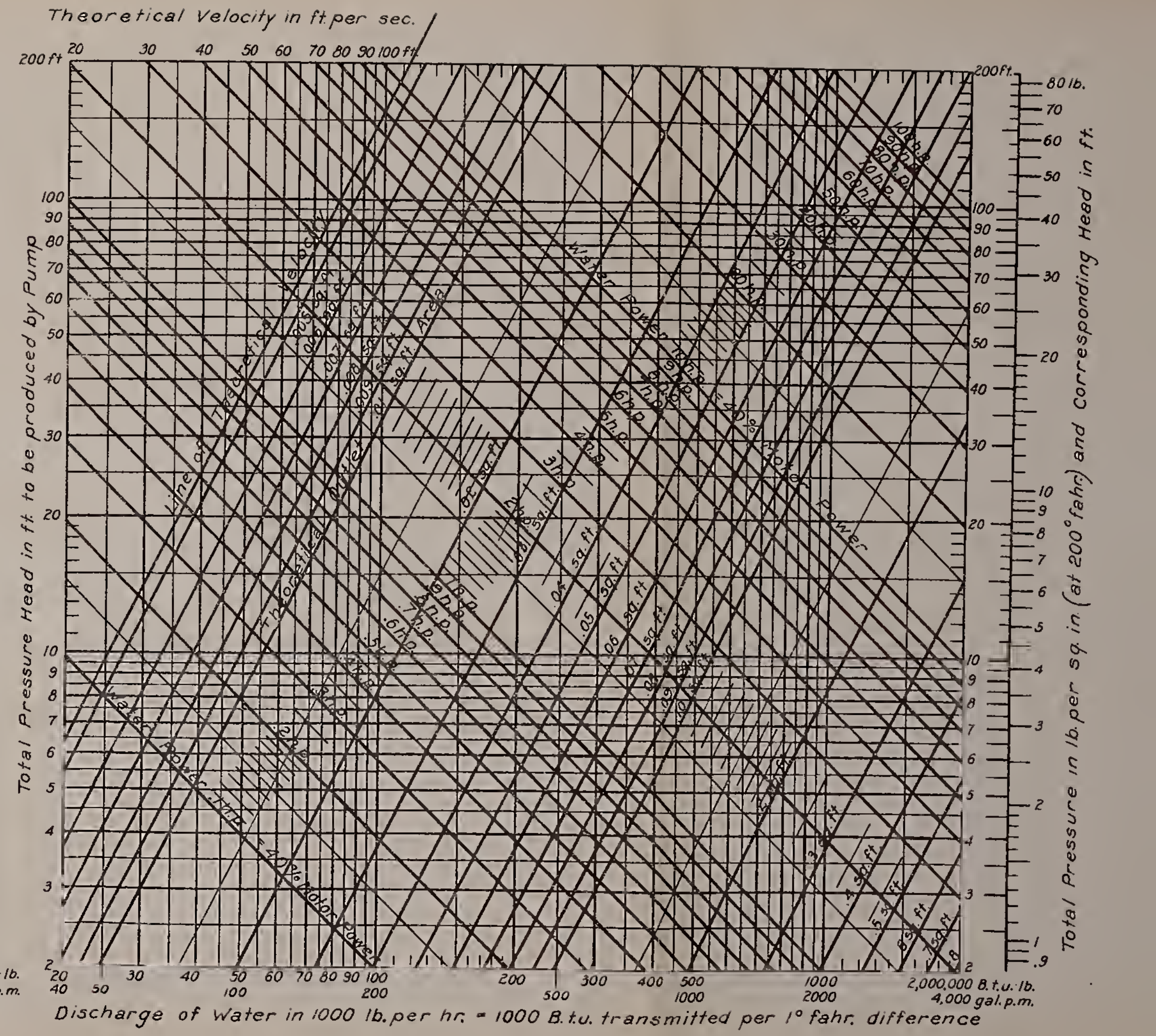
insulated pipes in conduits per sq.ft. of surface	50 B.t.u. per hour.
" " " buildings per sq.ft. of surface	100 B.t.u. per hour.
bare pipes in conduits underground per sq.ft. of surface	150 B.t.u. per hour.
" " " buildings per sq.ft. of surface	350 B.t.u. per hour.

less the energy to overcome friction and obstruction in B.t.u. per hour.

Corrections

If the heat absorbing and emitting points are on different levels, the head to be produced by the pump should be corrected according to differential weight and height. For water at 200° F., $H_c = H \pm .00037\,(t - t_1)h$ (approximate).





feet of water column into the corresponding pressure in pounds per square inch.

Distribution on a Circuit.—Since the flow of water to each individual radiator, coil or other appliance on a circuit must be governed by the head available for it, which is the differential between flow and return mains at respective junctions, it is evident, that the branches should be so proportioned that their resistance will equal this available head, when delivering the requisite volume. Between the back pressure in the flow system and the suction in the return there is a neutral zone in each branch circuit to and from the individual heat emitting appliances. In a closed system these zones, or points where neither suction nor excess pressure maintains, are simply under the head due to the water column above, which is balanced and does not bear on the flow. Hence the total difference of pressure, or loss of head,

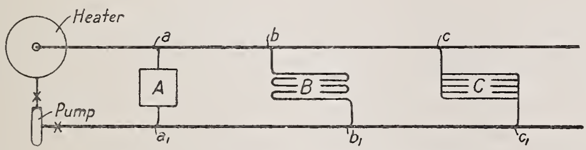


Fig. 6.—Schedule of forced hot water heating.

must be the same between the pump and any of the neutral points on the branches of delivery. It follows, also, that the head lost between two opposite junctions of flow and return mains must be the same through any of the individual connections. The idea is illustrated by Fig. 6, wherein the loss for the run from a to a_1 is equal to the total for the run a–b–b_1–a_1 and the total for a–b–c–c_1–b_1–a_1. Appliances nearer to the pump will therefore have a greater head available for the flow through their individual branches, and should be proportioned accordingly. For instance, the differential head between a and a_1 is greater than that between b and b_1. If sized for the same head, there will be an excessive delivery through a–a_1, or a partial short circuit. Under a constant head, this will mean a shortage for other branches toward the far end of the line. In other words, there will be uneven distribution, or, when adjusted by throttling, it will be distributed at a disadvantage, with pipes larger than necessary in some parts, and probably scant diameter in others, involving more friction. A pipe system that is adjusted by trial after installation is likely to cost more and to require more power,

thus increasing both the investment and expense of operation. Distributing lines designed to equalize the loss of head as nearly as is possible with the commercial sizes will be the most economical and may be relied upon to give the desired delivery. That the theoretical requirements can be closely approached by standard piping will be shown in application of Chart I.

To equalize the losses of head for the same total presupposes a fairly accurate estimate of the quantities of water to be carried in all parts of the conduit system. These quantities are not always proportional to the amount of heat to be delivered at certain appliances, but are often materially increased by the heat losses in transit. If we consider the latter as part of the heat to be carried, the volume of water to be moved increases at the same ratio.

As a general rule, the heat emission along the pipe lines should be estimated for all branches and for such portions of mains in which they would be a material item. This is usually done on hand of a schedule of tentative sizes, as will be outlined by the example to follow. When such losses occur in conduits branching into two or more individual lines, the latter must handle their proportionate shares of the excess allowed for the heat losses on the main. If the losses in the mains are relatively small, and partially offset by the heat developed through friction, it is proper to disregard the mains, inasmuch as the mean temperature and heat delivery can be kept up by a slightly greater range, without appreciable effects on the heat distribution.

Effect of Throttling.—A system of conduits should be designed for the maximum delivery and equalized with all branches carrying the full volume. Even then the distribution will be imperfect when any part of the system is shut off. The degree to which the discharge through the remaining outlets is affected depends upon the design and proportions of the conduits, also to an extent upon whether the speed of the pump or the power are under control.

If the power put into the flow is constant, the product $W \times H$ must remain the same. Thus, with the amount of water reduced by throttling, the total head is increased, and to a lesser extent also the delivery to the other branches. The pump will speed up automatically to produce this additional pressure, unless controlled to prevent it.

Under constant pump speed the total head H will remain practically the same with throttled discharge. It is effected

only in so far as the pump efficiency may vary under throttling. The volume in the main is naturally reduced according to the branches shut off, and therefore also the pressure losses along the line, which will leave a greater balance for the open branches in which the flow is increased. This increase, however, is not proportional to the extra pressure available, but only about to the square root of the same. Hence there will be a reduction in the total volume delivered. The power need at constant speed therefore decreases with the output or the product WH. The energy saved depends upon the efficiency of pump and motor under changing load.

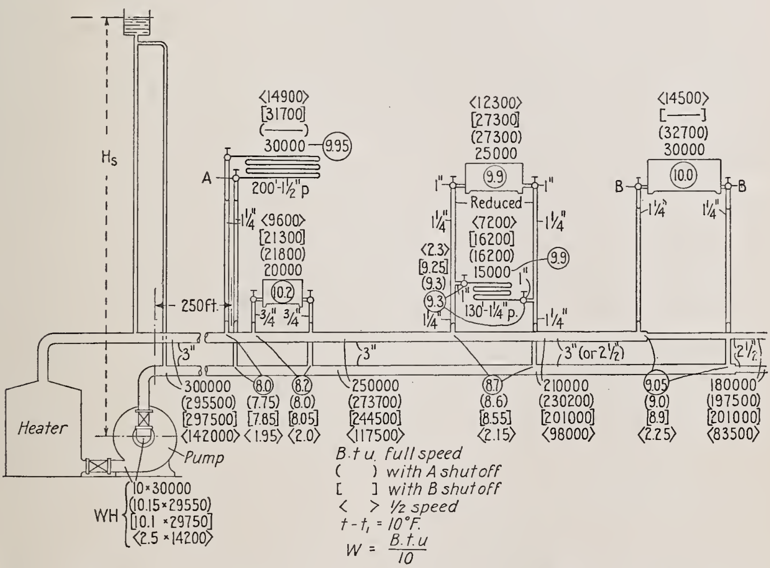


Fig. 7.—Example of forced hot water heating system showing effect on distribution of throttling and speed reduction.

Fig. 7 may serve as illustration of the effect of reduced volume on distribution. It is assumed that the pump speed is not under control and that the power put into the flow is constant. When the first branch A is shut off, it will be noted that the total delivery as marked in parenthesis (295,500) is smaller, and the total head (10.15) larger than with the last branch B cut off, for which case the figures are given in brackets [297,500] and [10.1]. If the volumes are calculated at which equalization will take place, it will appear that in the first case the excess volume

is about equally distributed through the balance of the system, while the throttling at B gives an appreciably greater increase to the nearest outlet ahead of it. It becomes evident also, that with relatively small pressure losses in the main, the total volumes and the distribution are less affected by the throttling of discharge outlets. In this particular instance, the main, if proportioned simply according to the volume carried (or W instead of $W \times H$) would be reduced to 2 1/2 in. for 210,000 B. t. u. This would make equalization more difficult with all outlets in use and would shunt practically the whole excess volume under curtailed discharge through the branches at the near end. Larger mains permit the initial pressure to be kept up well to the far end of the line and assure delivery under a greater variety of conditions. The velocity head as an element of uncertainty in distribution is at the same time kept down, and the trunk line approaches the idea of a reserve tank or plenum with perfect equalization.

Aside from the questions of power, first cost and heat losses, the sizing of a main, should, for these reasons, be governed by the relative importance of equal distribution. The extent to which delivery is to be throttled, and future outlets to be provided for should also enter into consideration.

Distribution under Reduced Head.—The bearing on distribution of a reduced total head, through slowing down the pump, is very slight. The volumes discharged through small branches are decreased somewhat more than those flowing through the larger ones, as shown again on Fig. 7, by the B. t. u. and losses of head given for half speed. This fact is due to the relatively greater head required for the flow in small pipes at low velocities. This will be true of course, only while the velocities are still above the critical stage below which the coefficient of friction is much reduced. Under ordinary conditions, the total delivery is less than that proportional to the speed of the pump. At half speed as shown, the discharge will be smaller than .5W, while the head created by the pumps cannot exceed $.5^2H$, unless through chance or poor selection its efficiency is improved under light load. The theoretical power decreases also as the volume, and is therefore smaller than $.5W \times .5^2H$, being a function of $.5^3 \times W.H$, assuming that the pump efficiency remains the same. If the latter decreases materially at lower speed, the actual power is liable to vary less than the cube of delivery. The

example on Fig. 7 indicates, through the <B. t. u.> calculated for half speed, that in ordinary practice the effect on distribution of slowing down or speeding up is negligible. The reductions in the volume, however, will be appreciably below the proportionate speed, especially when the effective total head is further decreased by lower pump efficiency. This fact should be borne in mind when speed reduction is to meet a stated requirement.

Heat Developed by the Flow.—When determining the heat losses in transit, it will pay in some cases to figure out and deduct the heat developed by the friction of the water in the pipes. This item may, under extreme conditions, result in higher temperatures at the far end of the flow main. The emission on the run remains practically the same, but the heat thus supplied, or returned to the system, reduces the necessary allowance in volume to make up for the losses in transit, especially for the mains, in which the heat losses are relatively smaller, while the friction is usually greater. This heat is the thermal equivalent of the mechanical power imparted to the water, and converted, as resistances are overcome along the pipe line. Expressed in B. t. u. per hour it is $= \frac{2545 \text{ W. H}}{3600 \times 550} = .00128$ W. H. Strictly taken, this heat should be figured for each section of conduit, subtracted from the losses by convection, which can be estimated, but for practical purposes it will suffice to establish the total, and its probable effect on the temperature of the flow. The value for H in this formula represents the sum of friction and resistance heads and should not include the dynamic head maintained. The latter manifests itself as *vis-viva* or mechanical power, until the pump is stopped, when the momentum of the flow is gradually lost in friction and only then converted into heat. It should include, however, the hydraulic losses in the pump, which can be estimated from its efficiency.

Taking for instance a tax on a system of 6,000,000 B. t. u. to be transmitted at 20° F. differential, and requiring 300,000 lb. of water per hour or 600 gal. per min., we find that this volume flowing through a 6-in. main at a temperature of 200° F., will lose about 8650 B. t. u. per hour by convection per 100 ft. run through an insulated pipe in a conduit. The resistance for that length, including four elbows, two right angle tees, and two gate valves, will take a head of about 7 ft. Hence the energy expended to overcome it would be $.00128 \times 300{,}000 \times 7 = 2700$ B. t. u.

per hour. This heat may be increased by other local resistances upstream, such as a coil, and will benefit more or less by the friction head of the pump, which generally doubles H, so that a moderate increase of speed, or better protection from heat loss, might easily bring the B. t. u. generated up to the heat losses in transit. The latter is therefore, most likely to become negligible as an item affecting the volume or the temperature. Inasmuch as the heat loss by convection along the mains is relatively smaller, while the resistance is relatively greater than that in branches, it seems quite proper under similar conditions to make no allowance to the volume of water to cover the heat losses in mains.

Temperature Head.—If the heat absorbing and emitting points are at different levels, the pressure to be exerted by the pump may be modified appreciably. Such differences of level between boiler and radiator will create a certain head, due to the unequal weight of the two water columns represented by the flow and return mains. This temperature head is proportional to the difference in weight per cubic foot and to the height of the columns and can be expressed by $h_t = h\frac{w_1 - w}{\frac{w + w_1}{2}}$, or by the variation in density for 1° F., which is also the coefficient of expansion. At about 200° F. $h_t = .00037(t - t_1)h$ (approximate). If heat is delivered at a higher level, the head due to differences in temperature will naturally accelerate the flow, unless it is discounted or subtracted from that produced by the pump. The reverse will take place when the heater is elevated above the cooling surfaces, in which case the temperature head is negative and must be added as a loss to the total to be overcome by mechanical means. The corrected head may be expressed therefore as

$$H_c = H \pm .00037(t - t_1)h$$

For transmission of heat at long distance, such differences in level have little bearing on the hydraulic slope, and on the flow of the water, the temperature head being only a small fraction of the total required to overcome the high resistances conditional in such cases. For individual heating apparatus where forced circulation is used mainly to accelerate, the ratio $\frac{h}{l}$ will be materially affected, especially with greater range of tempera-

tures. The corrected heads must be applied, of course, to each branch individually, when determining its size. Where the temperature range is varied according to weather, such a system should be balanced for average conditions and proportioned so that the extra head due to highest flow temperature will not materially affect distribution.

Static Head in a Forced Circuit.—In a forced circuit of a given height of water column (see Fig. 7) represented by the elevation of the expansion tank above the pump, there is a limit to the pressure difference that can be maintained by the pump. If the suction at the inlet overbalances this natural static head, H_s which would maintain at the state of rest, there will be an actual vacuum at that point. This vacuum will lower the boiling temperature of the water and may thereby generate steam at the pump suction, thus interrupting the circuit. When there is any danger of such occurrence, the tank level should be raised, or the resistances eased. In locating the tank it is best to place it at some neutral point along the circuit, or provide a neutral point by properly sized flow and return pipes to and from the tank. These pipes should present the same resistance to the flow as all other branch circuits when passing a sufficient amount of water to make up for the heat loss in tank and connections, thus preventing a short circuit near the pump.

Application of Chart I.—In order to apply the theories outlined, it is essential, in the beginning, to establish all the factors that may bear on a problem. The methodical way to do this is to present the situation graphically, giving a comprehensive view of the apparatus to be calculated. This may be obtained simply by a tracing or print of a plan of conduits, with elevations worked into it and special features noted. It should provide a schedule of lengths and heights to scale, show all items of local resistance, and permit the quantities carried to be marked down for all parts of a circuit. These quantities, in B. t. u. per hour or in gallons per minute, should be stated for each section of main and branch carrying a constant volume or weight, the total for all the branches making up the tax on the pump.

On the basis of these quantities the system may then be sized tentatively for an even rate of pressure loss. The initial velocity in the main is assumed as experience may dictate, according to the power available, length, or other features. Assuming for

instance a velocity of 7 ft. per second to carry 24,000,000 B. t. u. at 20° F. drop, or 2400 gal. per minute in a 12-in. pipe, the loss of head by friction will take place, according to chart I, at the rate of .15 ft. per 10 lin. ft. of length. The proportionate sizes for decreasing quantities, as branches are taken off, may then be found quickly along the same horizontal line of friction head. Thus a portion of main carrying 5,000,000 B. t. u. or 500 gal. would call for a 7-in. pipe as the nearest commercial size, giving a velocity of 4.2 ft. per second. For 40,000 B. t. u. it will be a 1-in. pipe, at about 1.5 ft. velocity. This method of proportioning gives a rational way of reducing speed for smaller sizes and permits a preliminary estimate of the total loss, which it may be necessary to keep down or to increase for various reasons. In the former case it is proper to choose a line of lower friction head by which to proportion the mains, or *vice versa*.

From these preliminary sizes the heat losses in transit should be estimated and added to the quantities. As stated, it is usually sufficient to allow for the losses in branches and bare portions of mains, but it must be remembered, that the increased volume for a branch must pass also through the mains, and through any further branches beyond.

When all quantities have been corrected, and allowance made for any temperature head, the final calculation of sizes with a view to equalizing can proceed on a safe basis. Starting at the pump, the losses of head by friction and local resistances, as derived from the chart are added together for the first section carrying the full volume up to the nearest branch. The same is marked on the schedule, just ahead of the junctions on flow and return mains. In using the chart, it is convenient to figure first the friction head for the length in question. This gives the velocity line along which the resistance head for elbows, T's, and other items of obstruction may be read. The process is repeated for the runs of mains to the next set of branches, and the sum of these being added to the first figure. The total at the last pair of junctions, plus the resistance of the branches and the appliance furthest from the pump will give the total pressure head for which all other branches nearer to the pump should be equalized.

The process of equalization is necessarily tentative, but with some practice and judgment in assuming the sizes, it will be found that one or two trials will suffice to find the nearest com-

mercial diameter, or combination of diameters, that will produce the desired resistance. As stated previously, there are various means of bridging the wide gaps in capacity between standard sizes. As a matter of fact, equalization may be carried to any degree of accuracy that may be desired. No valid objection can be raised to the use of different sizes for flow and return pipes, nor to changes in the diameter for any length desired. In such cases the items of head are simply obtained on two different velocity lines for the same volume. Frequently, the length of branches may be varied, or the flow and return mains looped in opposite direction. Practice will teach the best means for various situations.

The total head for a system, as determined for purposes of equalization gives also the data for sizing the pump and motor on hand of the auxiliary chart. The intersection of the line for the total head with that for the total volume carried, gives both the theoretical horse power and the equivalent orifice. The corresponding velocity is found at the intersection of the pressure line with that of theoretical velocity. The application of this data is explained in the general chapter on the flow of water.

Example.—The example presented by Fig. 8 shows a portion of a district heating system, with underground flow and return mains, insulated and laid in conduits. The water is set in motion by a centrifugal pump, heated by steam coils in a large drum through a range of 20° F. and conveyed to different appliances. Each of these is absorbing the heat through coils of pipe arranged in continuous circuits, as a part of the pipe system, or as a subsidiary individual heating plant, controlled by valves. The length and sizes of these coils, as far as they bear on friction and resistance, are stated on the schedule. The quantities are marked in B. t. u. per hour for each sectional run and the head is figured for each set of junctions. The flow and return mains being parallel throughout, it can be assumed that they present the same resistance since the slight reduction of velocity and friction in the return system due to the shrinkage is a very small item. The odd resistances of the apparatus near the circulating pump are figured separately. They include the item for a Venturi meter, for which a coefficient of resistance of .1, based on the throat velocity, has been assumed for safety, with a liberal allowance for friction. The losses of head for the heater and pump connections are added and set down as initial head for the mains. The figures given at the successive junctions accord-

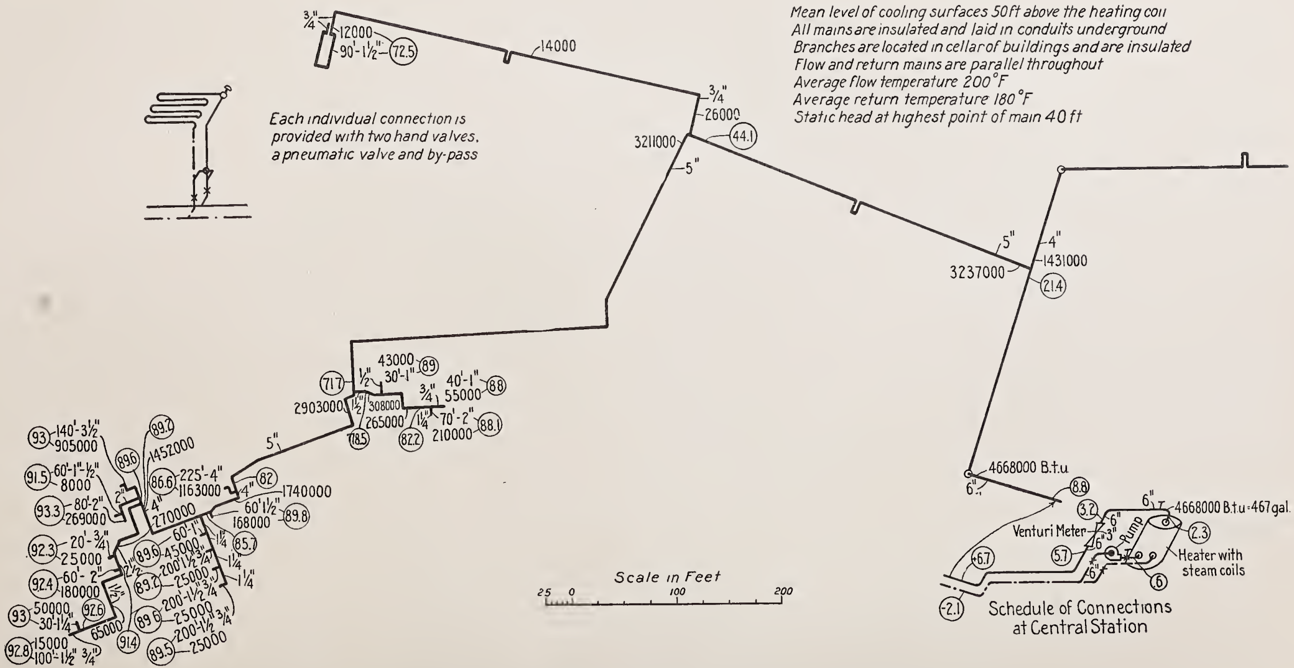


FIG. 8.—Example of forced hot water circulation for district heating.

ingly give the combined losses up to these points. The differential between these junctions must make up the total pressure exerted by the pump and govern the delivery for each unit on the circuit. The sizes of individual connections are relatively greater as the available pressure becomes smaller near the end of the run. This is specially noticeable at the first branch, supplying a small building at very long distance from the main. The excess of head at that point is fully 45 ft. and permits a surprisingly small connection, considering its length. On account of the long run of this branch the heat loss in transit is added to the tax, so that it will in reality receive more water in proportion to the B. t. u. to be delivered at the end. The branch has been kept somewhat larger than it figures, but the excess of water due to that cause is so small that it would hardly affect the delivery to the other appliances.

The differences in level do not exceed 50 ft. in either direction. The effect on the flow for a run of 1500 ft. accordingly figures $.00037 \times 20 \times 50 = .37$ ft. or less than 1/2 per cent. of the total, and is therefore a negligible quantity.

The correction to the volume for heat loss in transit is also very small, since the heat developed by friction practically offsets the former. At full speed the drop in temperature near the extreme ends is, in fact, barely perceptible according to actual readings.

With the exception of the first branch, already mentioned, the equalization is carried out to within about 5 per cent. of the total head. The discrepancies between individual branches are relatively greater, but the excesses and deficiencies of volume due to these faults in equalization are of necessity smaller again, and the effect on the heat delivered is still less. At any rate, the calculation is within the limits of error in sizing up the factors and arising from the uncertainties of execution.

The total head to be created by the pump may safely be taken as equal to the average calculated for the branches, in this case about 91 ft. or a pressure of 38 lb. per sq. in. According to the auxiliary chart this corresponds to a theoretical velocity of 77 ft. per sec. an orifice of .014 sq. ft. and ports of .15 sq. ft. area. The peripheral speed of the pump wheel should be between 85 ft. and 90 ft. per sec. and the blades curved to a small angle with the tangent to convert this motion into pressure as far as practicable. The sizes of the ports need not necessarily cor-

respond with those of the circulating mains. In fact, a hydraulically more advantageous arrangement is made with a pump having small outlets, say 5 in. diameter, connected to the 6 in. mains by easy taper pieces.

The power to overcome friction and resistances in this instance figures 11 h. p. by chart. The motive power provided is about 22 h. p. This assumes an efficiency of 50 per cent.

CHART II

HOT WATER HEATING BY GRAVITY

200° F. to 160° F.

THROUGH STANDARD WEIGHT IRON PIPES

Weight of water circulated in lb. per hour $W = \frac{\text{B.t.u. per hour}}{t - t_1}$

Velocity of flow in ft. per sec. $v = \frac{W}{3600 \frac{w + w_1}{2} a} = \frac{W}{3600 \times 60.56 \times a}$ for 180° F.

Head in ft. to overcome friction $h_f = .0257 \frac{v^{1.86}}{2g} \times \frac{l}{d^{1.25}}$.

" " resistance of obstruction $h_r = 1.38 r \frac{v^{1.86}}{2g}$.

" due to velocity $h_v = \frac{v^2}{2g}$.

Total head in ft. required $H = H_f + H_r$

Effective height, or actual height $\left\{\begin{array}{l}\text{available} \\ \text{and equal to } eH = \frac{\frac{w + w_1}{2}}{w - w_1} H \\ \text{required}\end{array}\right.$

with water columns at 200° F. and 160° F. $eH = \frac{60.56}{60.97 - 60.11} H = 70.8H$

or eH = difference in level between heat receiving and emitting surfaces.

The B.t.u. transmitted should include the heat loss in transit, as far as affecting the volumes for the runs to be figured. To estimate this loss take roughly for

insulated pipes per sq.ft. of surface	85 B.t.u. per hour.
bare pipes furred in per sq.ft. of surface	170 B.t.u. per hour.
" risers in rooms per sq.ft. of surface	255 B.t.u. per hour.
" horizontal pipes per sq.ft. of surface	340 B.t.u. per hour.

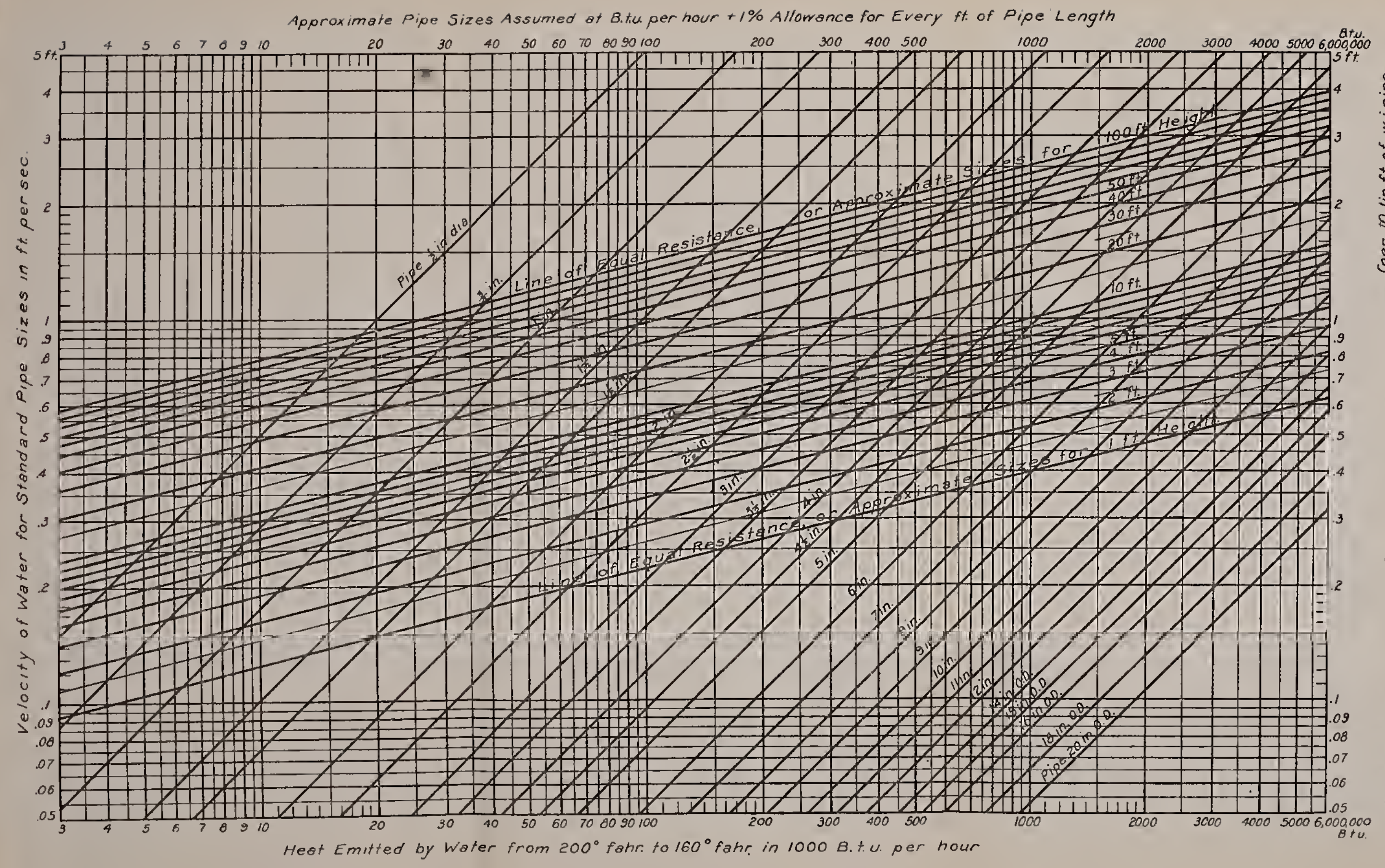
Corrections

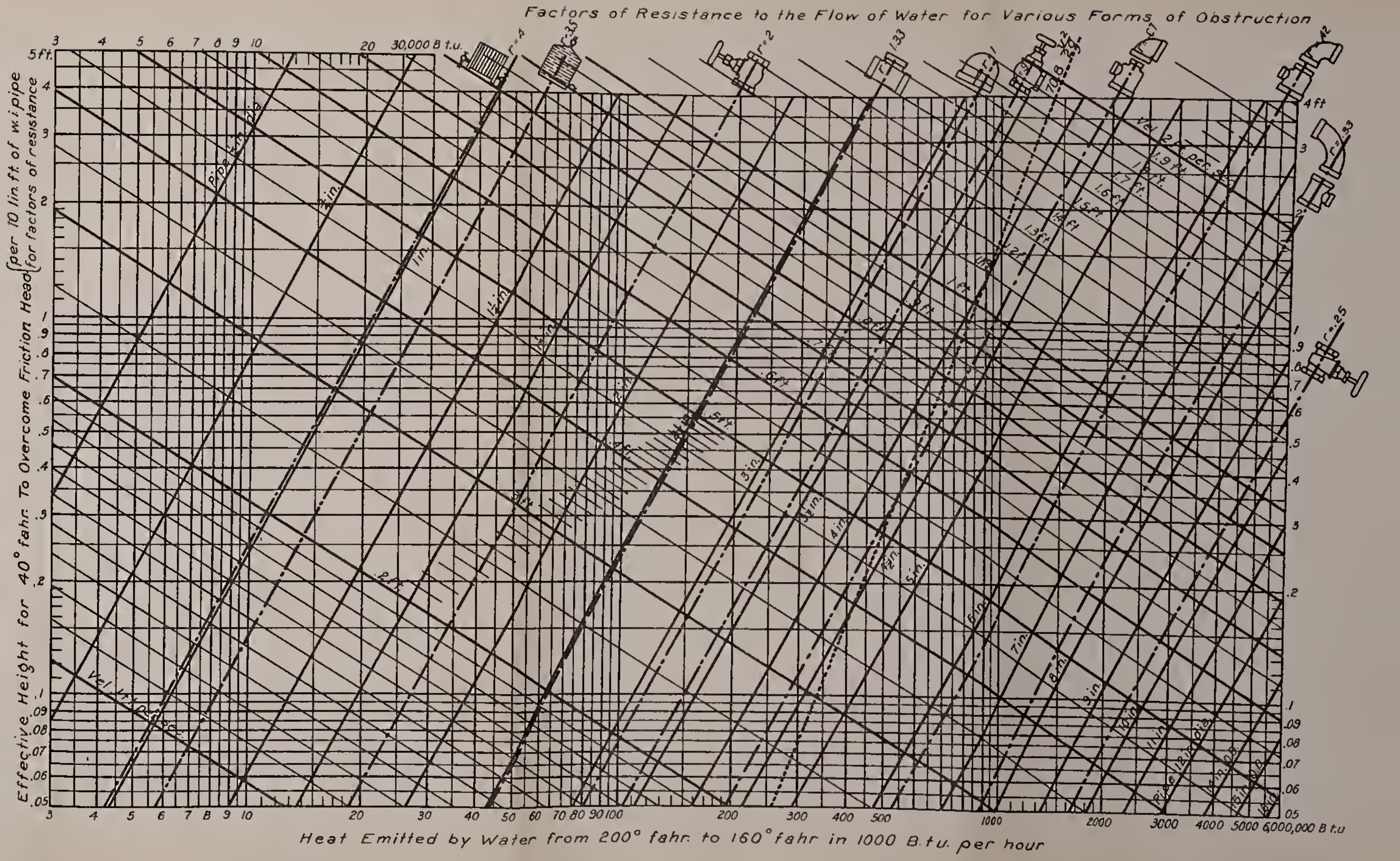
The range of 40° F. may be considered that between flow and return mains near heater.

If the heat losses are considerable, owing to long runs or bare piping, the emission in transit should be figured for the circuits affected, and the mean level of heat emitting surfaces established. The resistance head for the increased B.t.u., or water volume, should be within these mean effective heights.

If a system is to be calculated for a greater or smaller range throughout, the factors below are to be used.

For $t - t_1$ =		36°F.	multiply B.t.u.	and h	by	1.11	at about 180°	mean temp.
"	"	38°	"	"	"	1.05	"	"
"	"	40°	"	"	"	1.00	"	"
"	"	42°	"	"	"	.95	"	"
"	"	44°	"	"	"	.91	"	"
"	"	46°	"	"	"	.87	"	"
"	"	48°	"	"	"	.83	"	"
"	"	50°	"	"	"	.80	"	"





CHAPTER IV

HOT WATER HEATING BY GRAVITY.

Charts II, III and IV.—Gravity circulation may be effected under widely different conditions. Hygiene demands liberal heating surfaces, at temperatures considerably below the boiling-point, while economy tends in the opposite direction, toward smaller surfaces and piping. Aside from structural reasons that may also enter into the problem, the relative importance of those two points will decide whether an apparatus should be designed to meet the extreme tax at moderate temperature, or whether it be permissible to depend on greater intensity of heat, and a greater drop, in order to reduce the expense. The diagrams Nos. II and IV may be said to be within the limits of good practice in either direction. It will not pay, except in special cases, to figure on lower flow temperatures than 180° F., on account of the rapidly increasing heating surface, nor on a smaller drop than 30° F., since the bulk of water becomes too great and takes too much time for reheating. To use a greater range than 40 degrees at 200° F. flow temperature, will make the piping undesirably small in some places, and involves closer calculation. The third diagram represents average practice. The choice in each case is determined in general, as stated, by considerations of hygiene and economy, but is also influenced by structural conditions and the necessity of quick reheating.

Closed systems, for medium and high pressures, which owe their earlier development to the facility of forcing, are gradually becoming obsolete in the same measure as the means of obtaining perfect circulation, under all sorts of conditions, with an open system at low temperatures are better known and appreciated. For this reason no diagrams have been worked out to fit their case. A closed system can be calculated, if need be, on hand of the present charts with proper application of corrections.

For apparatus which depends on artificial devices for accelerating the circulation by steam or other fluids of lesser density in one form or another, special charts might be worked out to

cover the theories in question. In view of their limited field of legitimate use and on account of the great variety of systems and schemes, each presenting a different problem, no attempt has been made to cover these special cases. They are applications of the same general principles and a study of the forces active in natural circulation gives the best means, and is necessary in fact, for planning artificial circulation to best advantage.

Balanced Gravity Circulation.—It is assumed that the heating requirements to be met by the different appliances, radiators or coils have been determined, and that these heat emitting surfaces are disposed so that they will deliver the thermal units as intended. The problem will then be to distribute the heat carrier accordingly. In individual plants sufficent power is usually

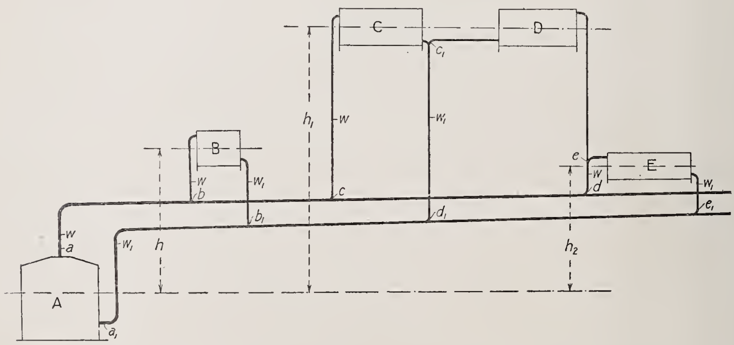


Fig. 9.—Schedule of hot water heating system with underfeed mains.

created for moving the required amount of water by the act of warming, through which a very small portion of the heat applied is liberated as live force in expanding the fluid. This hydromotive force is given by the differential weight between the warmer and cooler columns, usually designated as flow and return. It is dependent on temperature difference and on the height between the mean levels of heat absorbing and emitting surfaces, as expressed by $h(w_1-w)$, using the designations on Fig. 9.

At even distribution to each unit, suited to its cooling capacity, and without heat losses in transit, the differential density w_1-w will be the same throughout a system. The height h on the other hand, will differ, being given for each radiating unit by its level above the boiler, which level can be varied only

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CHART III

HOT WATER HEATING BY GRAVITY

190° F. to 155° F.

THROUGH STANDARD WEIGHT IRON PIPES

Weight of water circulated in lbs. per hour $W = \frac{\text{B.t.u. per hour}}{t - t_1}$.

Velocity of flow in ft. per sec. $v = \frac{W}{3600 \frac{w + w_1}{2} a} = \frac{W}{3600 \times 60.70 \times a}$ for 172.5°.

Head in ft. to overcome friction $h_f = .0257 \frac{v^{1.86}}{2g} \times \frac{l}{d^{1.25}}$.

" " resistance of obstruction $h_r = 1.38 r \frac{v^{1.86}}{2g}$.

" due to velocity $h_v = \frac{v^2}{2g}$.

Total head in ft. required $H = H_f + H_r$.

Effective height, or actual height $\left\{\begin{array}{l}\text{available}\\ \text{and equal to}\\ \text{required}\end{array}\right.$ $eH = \frac{\frac{w + w_1}{2}}{w - w_1} H$

with water columns at 190° F. and 155° F. $eH = \frac{60.70}{61.065 - 60.340} H = 83.8H$

or eH = difference in level between heat receiving and emitting surfaces.

The B.t.u transmitted should include the heat loss in transit, for all flow branches, risers and connections, also for bare flow mains. To estimate this loss take roughly for

insulated pipes per sq.ft. of surface	80 B.t.u. per hour.
bare pipes furred in per sq.ft. of surface	160 B.t.u. per hour.
risers in rooms per sq.ft. of surface	240 B.t.u. per hour.
" horizontal pipes per sq.ft. of surface	320 B.t.u. per hour.

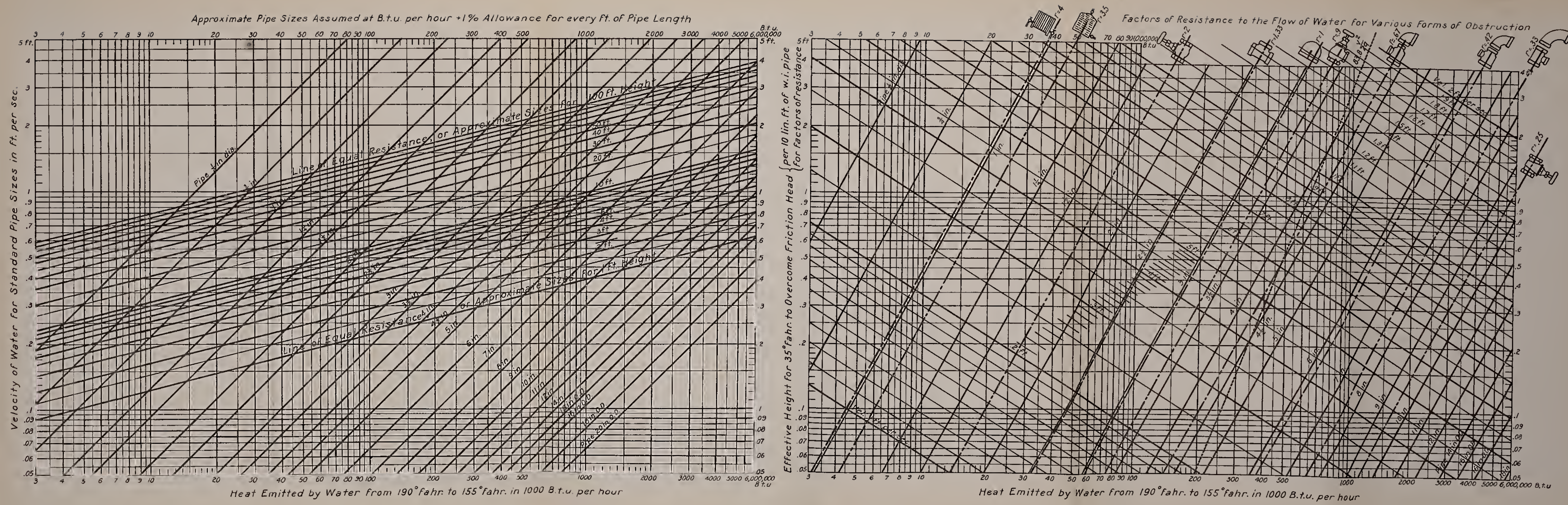
Corrections

The range of 35° F. may be considered that between flow and return mains near heater.

If the heat losses are considerable, owing to long runs or bare piping, the emission in transit should be figured for the circuits affected, and the mean level of heat emitting surfaces established. The resistance head for the increased B.t.u., or water volume, should be within these mean effective heights.

If a system is to be calculated for a greater or smaller range throughout, the factors below are to be used.

For $t - t_1$ =	29° F.	multiply B.t.u.	and h	by	1.21	at about	172.5°	mean temp.
"	31°	"	"	"	1.13	"	"	"
"	33°	"	"	"	1.06	"	"	"
"	35°	"	"	"	1.00	"	"	"
"	37°	"	"	"	.95	"	"	"
"	39°	"	"	"	.90	"	"	"
"	41°	"	"	"	.86	"	"	"



by raising or lowering the boiler and radiation, which is not practicable as a rule. With equal temperatures in the several columns of one system, it is the height, therefore, which determines the *pressure available* for creating the flow. This actual height, or the corresponding pressure, must be equal to the *pressure required* to overcome the combined losses by friction and local resistances $H_f + H_r = H$ for the entire circuit bounded by the respective appliances. Since the mean density of the water for the circuit is $\frac{w + w_1}{2}$, the pressure to be balanced by $h(w_1 - w)$ is equal to $\frac{w + w_1}{2}H$. This is in substance the fundamental equation used by Rietschel[1] and quoted in all standard books for calculating gravity circulation. It gives the value for $h = \frac{\frac{w + w_1}{2}}{w_1 - w} H$, which is the actual height of the water column required to produce the flow, under a certain drop in temperature. It is identical with what is usually called the *effective height* eH, and must be equal to the *height available* between mean levels of heat emitting and absorbing surfaces.

Distribution of the water to a number of appliances on different levels, and with varying lengths of piping, will hinge therefore upon the adjustment of the total loss of head along each circuit completed by an individual branch, to the pressure available for the respective points of delivery as given by the levels of the same relative to the heater. The pressure losses for the runs $a-b-B-b_1-a_1$, or $a-b-c-C-c_1-d_1-b_1-a_1$ or $a-b-c-d-e-E-e_1-d_1-b_1-a_1$ in Fig. 9 should therefore be made equal to the differential pressures created by the rising and falling columns of the heights h, h_1 and h_2, respectively. The principle may be followed with any scheme of piping, any number of branches, and for any distance and height, there being always a complete circuit traceable for each individual appliance.

The volumes of water governing the pressure losses in the mains are those actually carried by the respective runs, and the total up to any junction is a part of the sum identical for all branches beyond. The losses of head in the mains, therefore,

[1] "Theorie und Praxis der Bestimmung der Rohrweiten für Warmwasserheizung." H. Rietschel.

must always leave a margin of head available for the branches to the appliance having the least slope $\frac{H}{L}$, and the branches to those at higher levels or with shorter runs, must use up the surplus head, whatever it may be, by the resistances in rising lines and connections. *Instead of equalization* for the same total head as in the case of a forced system the *problem is to balance,* the unequal weights by conforming the resistance to the varying heights.

It might seem, that with an available head exceeding that required, each branch will obtain at least its share of the water. This would be true, if all branches were run separately, to and from the boiler. With a collective main, however, the total losses of head for the several branches on a main are interdependent. An excessive volume flowing through one branch will necessarily also increase the friction in the main and reduce the balance of head available for some other branch further along. If this branch supplies a radiator on a lower level, it may easily happen that no head is left for the same. The water will not circulate through such a branch until the differential weight, increased through cooling, will establish the balance again, but for a reduced flow, and reduced mean temperature at delivery.

Effects of Throttling.—The facility for central regulation which is peculiar to hot water heat, and is increased in range through accurate balance, eliminates to an extent the frequent necessity for shutting off individual radiating units. A pipe system should nevertheless be designed with a view to maintaining an equable distribution under such variations as are liable to occur through the temporary closing of appliances. The varying heads for the individual radiating units will naturally bear on distribution with curtailed delivery.

In order to compare the effect with that obtained under an even total head created mechanically, the example illustrated by Fig. 10 is chosen to represent the identical case of Fig. 7, but without the extra length of main which calls for the forced flow in the former instance. It is assumed also that the range of temperatures is from 180° F. to 150° F., instead of only 10° F. The head required for each radiating unit, when receiving its share of water, is marked thereon. The approximation to the head available is carried close enough to give the assurance

that the actual volumes and temperatures can deviate only by small fractions from those desired. The effect of shutting off the first branch A is shown again by the B. t. u. in parenthesis, ( ). It will be seen that on account of the partial self-regulating tendency of gravity systems, the increase of volume for the other branches is not nearly as great as with the forced system. The reason for this lies in the decrease of the differential weight $w_1 - w$ due to the excess volume of water passing through the branches left open. This decrease tends to reduce the effective head at the same ratio for radiators at any elevation, but the

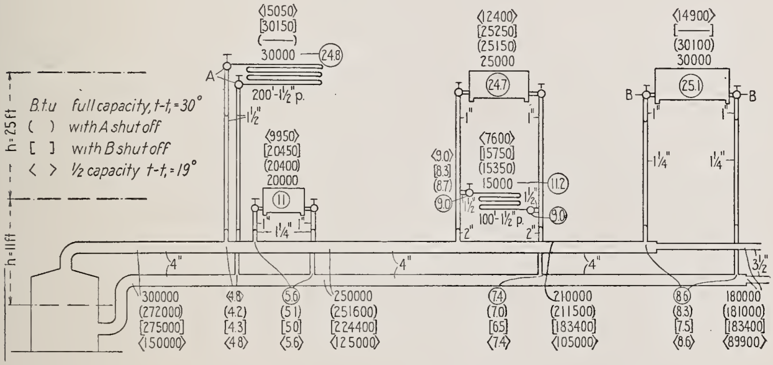


FIG. 10.—Example of hot water heating system showing effect on distribution of throttling and reduced capacity.

diminished friction head in the main leaves a proportionately greater height available for the flow to radiators on a lower level. The latter will therefore receive most of the excess volume, irrespective of the location of the branch cut off. This fact is confirmed by the figures in brackets, [ ], which give the B. t. u. with B shut off. We find for that case also a slightly greater excess for the branches ahead of B. The example shows, that with 10 per cent. of the heating surface shut off, the delivery to other appliances increases in no case more than 3 per cent., and the variation between them is less than 2 per cent. While other cases, with relatively smaller mains and larger risers, would not fare as well, it is evident, that distribution is not materially upset by throttling a considerable portion of the radiators, if a pipe system is properly balanced.

Distribution at Reduced Capacity.—At reduced load, that is, with lower flow temperature as should be carried to suit the

weather, but with all appliances in use, the balance of a system is not fully maintained in theory. The mean water temperature at which, say one-half of the heat emission at 165° F. will take place is about 120° F. The drop in temperature, with the full volume passing through, would then be only 15° F. With the differential weight and the available pressure $(w_1 - w)h$ thus reduced, the velocity will also diminish and bring the range of temperature up to an intermediate point, which can be found by tentative calculation. When the head available checks with the head required, the latter will give the velocity of the flow and the heat deliverable at each appliance, under these conditions.

These amounts of heat delivered are figured up individually for the same example. It will be seen that the deviations from the desired quantity, that is, from 1/2 B. t. u., are relatively small, and have practically no effect on the rate of heat emission. The variations as they appear are due again to the slight shifting of the friction losses, the smaller branches being somewhat under a disadvantage at reduced velocities. Conditions will be different again at very light load, as the velocities drop below the critical stage, which is reached first for the smallest diameters.

Higher temperature at the top of a flow pipe, which is a sign of even flow, as taking place below the critical velocity, also the variations of the coefficient of friction due to temperature, may sometimes upset the balance slightly or temporarily, but these phases, however they may affect distribution at very low temperatures, need hardly be considered.

Application of Charts II, III and IV.—The three charts for gravity circulation are based on the assumption of uniform difference of temperature $t-t_1$ between flow and return. The calculation is thereby simplified by a fixed ratio of $\frac{w+w_1}{2}$, to $w_1 - w$, and allows the charting of eH in place of h given on other charts. In other words, the losses of head by friction and obstruction are charted on a scale $\frac{w+w_1}{2}$ to $w_1 - w$ which is 100.3 to 1 for 180° F. to 150° F., 83.8 to 1 for 190° F. to 155° F., and 70.8 to 1 for 200° F. to 160° F., thereby giving directly the height of the two water columns necessary at the stated temperature range for overcoming resistances to the flow. The means are thus provided for immediate comparison of the head required with

CHART IV

HOT WATER HEATING BY GRAVITY

180° F. to 150° F.

THROUGH STANDARD WEIGHT IRON PIPES

Weight of water circulated in lb. per hour $W = \frac{\text{B.t.u. per hour}}{t - t_1}$.

Velocity of flow in ft. per sec. $v = \frac{W}{3600 \frac{w + w_1}{2} a} = \frac{W}{3600 \times 60.87 \times a}$ for 165° F.

Head in ft. to overcome friction $h_f = .0257 \frac{v^{1.86}}{2g} \times \frac{l}{d^{1.25}}$.

" " resistance of obstruction $h_r = 1.38 r \frac{v^{1.86}}{2g}$.

" due to velocity $h_v = \frac{v^2}{2g}$.

Total head in ft. required $H = H_f + H_r$.

Effective height, or actual height {available and equal to required} $eH = \frac{\frac{w + w_1}{2}}{w - w_1} H$,

with water columns at 180° F. and 150° F. $eH = \frac{60.87}{61.164 - 60.557} H = 100.3H$,

or eH = difference in level between heat receiving and emitting surfaces.

The B.t.u. transmitted should include the heat loss in transit, as far as affecting the volumes for the runs to be figured. To estimate this loss take roughly for

insulated pipes per sq.ft. of surface	75 B.t.u. per hour.
bare pipes furred in per sq.ft. of surface	150 B.t.u. per hour.
" risers in rooms per sq.ft. of surface	225 B.t.u. per hour.
" horizontal pipes per sq.ft. of surface	300 B.t.u. per hour.

Corrections

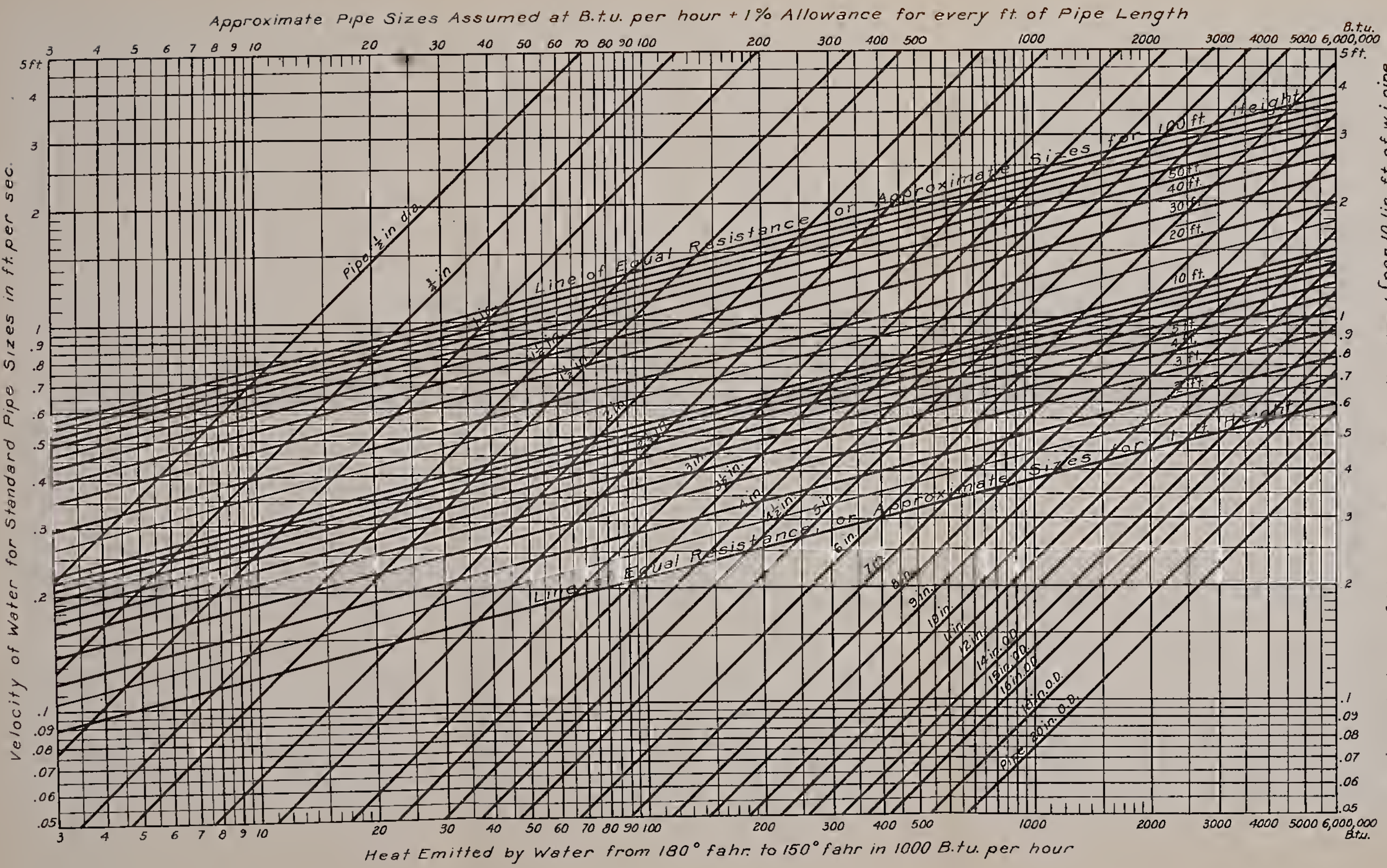
The range of 30° F. may be considered that between flow and return mains near heater.

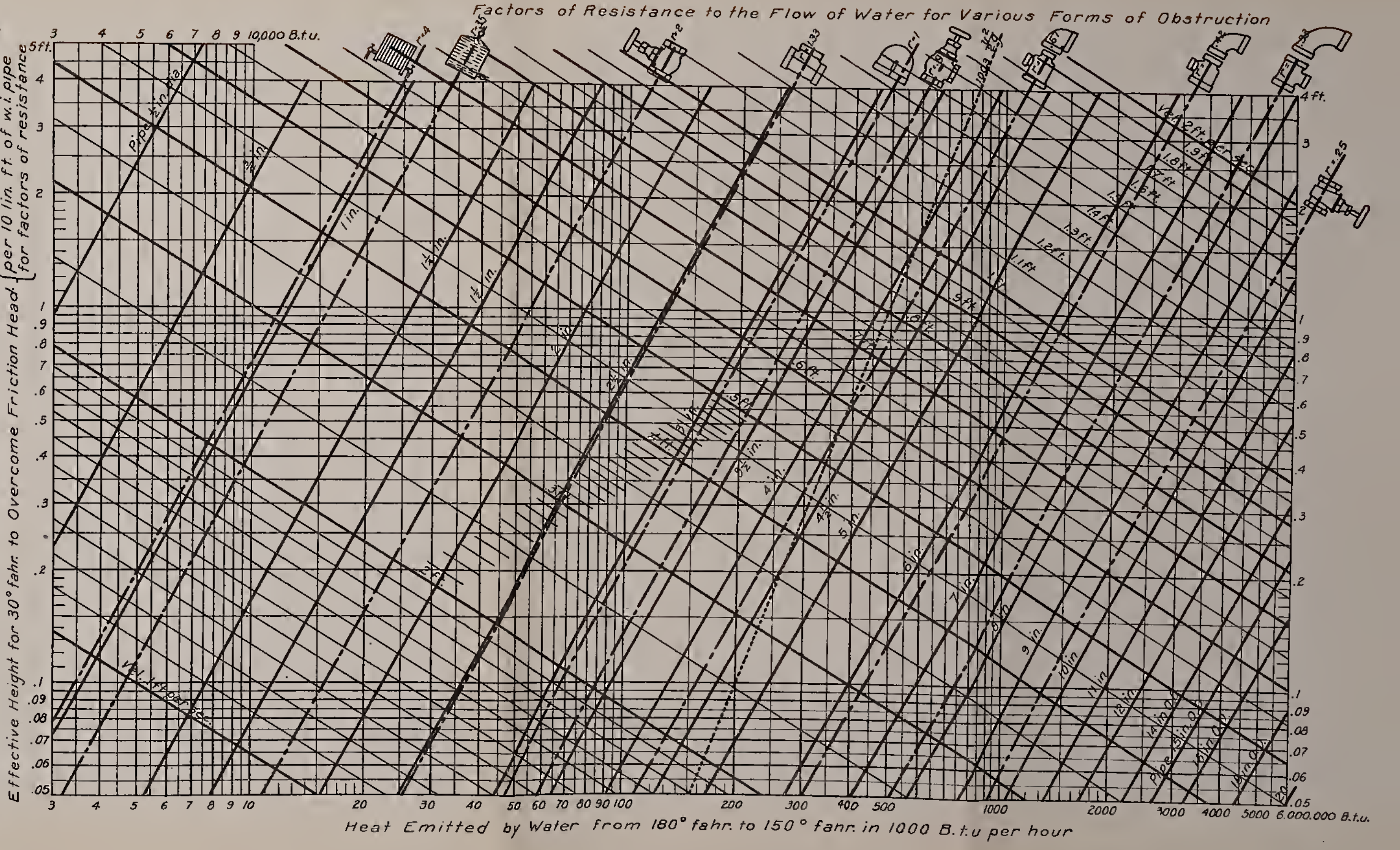
If the heat losses are considerable, owing to long runs or bare piping, the emission in transit should be figured for the circuits affected, and the mean level of heat emitting surfaces established. The resistance head for the increased B.t.u., or water volume, should be within these mean effective heights.

If a system is to be calculated for a greater or smaller range throughout, the factors below are to be used.

For $t - t_1 = 20°$ F., multiply B.t.u. and h by 1.50 at about 165° mean temp.

"	22°	"	"	" 1.37	"	"	"
"	24°	"	"	" 1.25	"	"	"
"	26°	"	"	" 1.15	"	"	"
"	28°	"	"	" 1.07	"	"	"
"	30°	"	"	" 1.00	"	"	"
"	32°	"	"	" .94	"	"	"
"	34°	"	"	" .88	"	"	"





the actual height available for any circuit, which is a decided convenience and permits a better grasp of the problem in hand.

In order to figure systematically, it is advisable to draw up a schedule of the entire piping as planned, showing horizontal and vertical distances to scale as illustrated by the examples to follow, and giving the tax on each branch and run of mains in B. t. u. per hour. Since the losses of head depend on the pipe sizes yet to be calculated, the latter must be assumed tentatively. This can be done by means of the charts for approximation which give the nearest commercial size for a given quantity in B. t. u., at any velocity, and the actual height necessary to overcome a uniform nominal resistance, assumed equal to that for a pipe length of 100 ft. The pipe schedule with heights and quantities can be sized directly from the diagonal lines representing about the mean level of the lowest surfaces, if a fair allowance is made at once for varying horizontal distances by adding about 1 per cent. to the B. t. u. for each linear foot of pipe length from the boiler. For a radiator to emit say 12,000 B. t. u. located about 75 ft. away, at 20 ft. elevation, it is advised, for instance, to add 75 per cent. The approximate size of a connection may then be found at the intersection of the vertical line for 21,000 B. t. u. with the line of equal resistance for 20 ft. elevation. For 30° F. differential, the next larger standard pipe will be 1 1/4 in. For 40° F., it would be 1 in. diameter. The risers and mains may be sized in the same way, by estimating roughly the mean length of piping for the total to be carried and taking as height the elevation of the lowest radiator on the line.

When the entire schedule is sized in this manner, the heat losses in transit should be estimated on hand of the data furnished. These losses are to be added to the tax in B. t. u. for the respective runs, to allow for an increased volume that will keep up the desired mean temperature. In so far as it emits heat, the distributing system should be considered as part of the delivering surface which defines the upper turning point, or mean level of cooling. If these losses in transit occur above or below the appliance, this level will be materially affected and call for certain corrections of the height in question, as will be further explained.

The pipe schedule will then give the volumes carried in any part, the assumed sizes, the lengths and the local items of obstruction, from which data all resistances can be figured accu-

rately by means of the other diagram. Starting at the boiler, the friction losses for each run between two junctions, and carrying the same volume, can be computed quickly from the scaled length, the chart giving the friction heads for 10 lin. ft. at the intersections of the lines for the B. t. u., and those of the assumed diameters. The several items of friction and local resistance for one section of constant diameter are found along the same velocity line and can be conveniently added up, the sum being noted at the junction. The total up to the next junction is added to the first, and so on successively, up to the points of delivery as will be illustrated by various examples. If flow and return pipes are parallel, the calculation can be made at once for both, the head to be stated being the sum of the two. With overhead feed, and wherever the return follows a different course, they are to be figured separately. The sum of the losses or the head required for the entire circuit can thus be established for some representative individual heat appliances in extreme location at lowest level and be compared with their actual heights above the boiler.

If the available head is used up before the connections to these units are reached, or if the branches at the extreme end become larger than seems practicable, the mains will have to be enlarged, beginning at such parts which show the highest rate of resistance. Sometimes the conditions will make preferable a general increase of the available head by lowering the boiler, particularly where a large water contents is to be avoided. A system with short horizontal runs will often give opportunity for reduction of sizes, both for mains and connections. Sometimes it will be found that individual branches to radiators at considerable elevation call for smaller sizes, theoretically, than seems desirable for durability and strength. In such instances it may be advisable to rely on adjustments by throttling, through key valves or other devices, but as a rule, it is best to reduce the size of the return branches for certain lengths, where least objectionable. Uneven sizes for flow and return, also reductions for odd lengths are indicated wherever the standard pipe sizes do not permit a reasonably close approximation of the available head. The losses of head for such portions of constant volume but uneven diameter are to be read off along the respective velocity lines, with the extra resistance of the reducing pieces added as items of obstruction. If the sizes are changed

by reducing elbows, the loss may be taken to equal that for a regular elbow of the smaller diameter.

It is good practice in general to bring the resistance head to within about 10 per cent. of the effective head available, keeping the former lower than the latter as a factor of safety. It would be waste of time to carry the equalization closer, inasmuch as the limit of error in figuring, measuring, and through the uncertainties of execution, will make further approximation of questionable value, and the discrepancies in the temperature of heating surfaces are, in any case, much smaller than those of the resistance heads. The heat delivery, therefore, is kept within the probable errors in estimating the requirements, and fair distribution may be counted upon for any reasonable flow temperature, thus assuring the perfect means of central regulation. The calculation in itself will show why a true balance is never obtained by rules of thumb, and usually remains imperfect if based on approximation tables or effected by artificial adjustment. Unless a gravity system is balanced by calculation on the basis of head available and resistance offered, the discrepancies are liable to be very much greater. At best, the pipe sizes resulting from approximate methods are always larger on the whole than they need to be. They generally require higher flow temperature before the circulation is established throughout and involve greater bulk of water. Hence the scientific method will not only assure accurate heat delivery, but it will reduce the time for reheating materially and make hot water heating apparatus more responsive.

The solution of these problems on a sound, physical basis will also show certain fallacies that have arisen through lack of understanding. For instance, circulation does not depend on pitch, so long as the air can escape. Again, small pipes may be used without fear of clogging, if the means are provided to flush them. Nor will they rust on the inside.

There is really no basis for the old notion that uneven sizes for flow and return should not be used. Delivery is governed by the resistance in the individual circuit as a whole, not by the smallest pipe size in any part of it. The variation in sizes for parts of the length of branches is a safe and practicable way of adjusting the resistances to the head available. The friction head in reduced connections can be foretold with greater certainty than the effect of any throttling devices. It cannot be

disturbed or tampered with, and saves the vexatious process of adjustment during operation.

The calculation as outlined may seem at first to require more time than is usually given up to such work when done by the popular approximation tables, but a repeated use of the charts will show the method to be practicable and more economical in the end. It is really based on the same principles as followed for years in doing accurate work by means of formulæ and extensive tables. The charts, however, give a more convenient means of application and the procedure suggested should induce a clearer grasp of each problem as a whole.

General Corrections.—As a rule, where higher temperatures are resorted to for the sake of economy it is also desired to use smaller piping. Aside from this, the assumption of smaller ranges for lower mean temperatures on the three charts is arbitrary. A pipe system computed by either of the charts will balance, practically, at any flow temperature, but the stated range will maintain only at the stated mean temperature.

The ranges of temperature, however, on which these gravity charts are based will be found to give considerable differences in pipe sizes, and it is desirable, occasionally, to assume an intermediate range where extreme length of runs and other items of resistance, or the lack of height available would call for larger sizes of mains than seems practicable or economical and where one of the other charts would lead too far from the desired results. In such cases a range of say 32° F. or 36° F. at the same mean temperature, will be indicated. The mean density $\frac{w+w_1}{2}$, remaining the same, the effective height $\frac{\frac{w+w_1}{2}}{w_1-w}H$ varies inversely as w_1-w, or the range of temperature. The volumes also are the inverse of the range. The B. t. u. on the charts are accordingly subject to correction by the factors giving the true volume, on the basis of which the resistances must be figured. The readings of the head are then to be corrected again for the corresponding differential head that is required to establish the balance. The calculation of a pipe system in other respects follows exactly the same course. It is not materially complicated, if the intermediate range of temperature is chosen to give a factor in even figures, say 10 per cent., plus or minus, so that it can be applied conveniently.

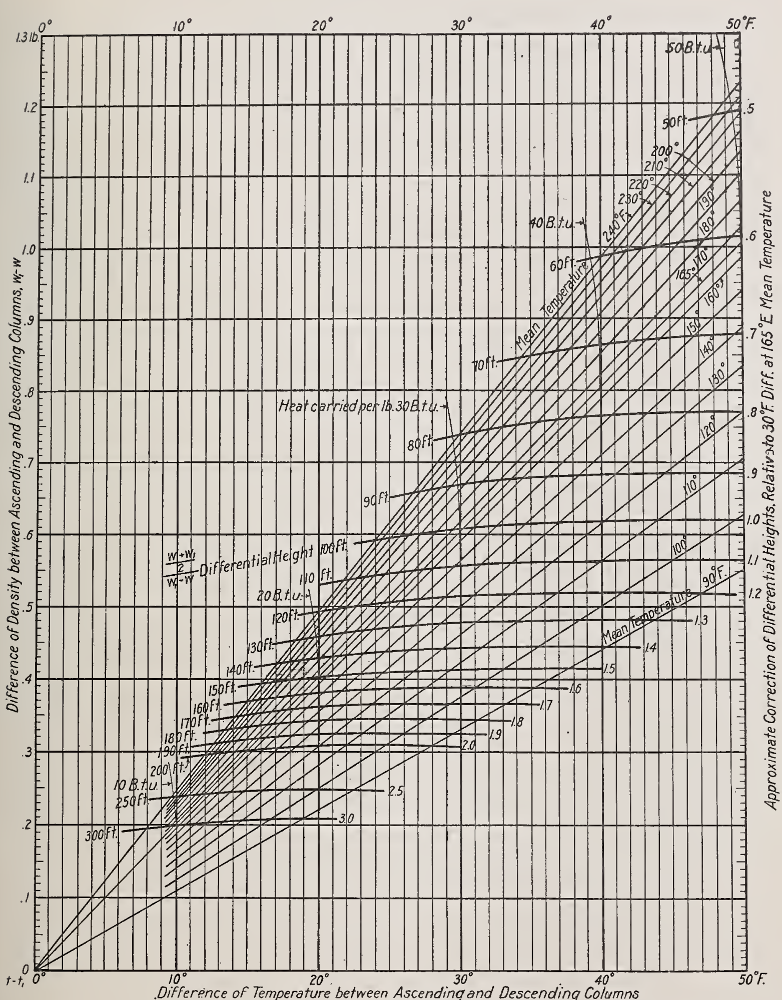


DIAGRAM B.—Table of differential heights for hot water heating.

The factors for correction are strictly applicable only to the stated mean temperatures, as the differential varies appreciably for cooler or hotter water, especially near to the boiling-point. The values given on the chart for 180° F. to 150° F. should only be used for mean temperatures near 165° F., and those on the other two charts for about 172° F. and 180° F.

For other mean temperatures the factor for correcting the head will have to be established from the table of differential heights, which is an extract from the table of properties, made for the purpose of ready computation of the hydromotive force available for a liberal range of conditions. It gives the differential density or pressure per foot of height, with the corresponding actual water columns, and the factors relative to the height of 100.3 ft. on which the chart for 180° F. to 150° F. is based. The table is specially convenient for the solution of problems in which the variations of power with different mean temperatures should be considered.

Using the table, we will find for instance, that the head for 25° at 190° F. mean temperature is 110 ft. that is, it takes 110 ft. differential water column to produce 1 ft. of actual head. The factor for correction is therefore about 1.1. At 165° F. mean temperature and 25° F. range it would be 1.2. For 45° difference, at 150° F. mean temperature, for another instance, the reading gives a factor of .73, while at 180° it would be .63. It is important to take into account these variations when dealing with higher temperatures generally, or with varying difference in one and the same system. At 220° mean temperature, for example, the same range of 30° results in a difference of weight of .72 lb. per cubic foot as compared with .60 lb. at 165° F. thus reducing the head necessary to balance it from 100 to 83 ft.

It is not often necessary to assume other mean temperatures for gravity systems, than those for which the charts apply, but it should be remembered in dealing with these odd cases, that the modified factors from the table apply, of course, only to the differential heads. The ratios on the chart for correcting the volume vary but slightly for other mean temperatures, and may be taken as constant in practice.

Individual Corrections.—As stated previously, the balance of a gravity circulating system will depend on, or should result in, a uniform drop of temperature between all the rising and the falling columns, but a strictly uniform difference for all sets of branches,

and for their entire height could only maintain if no heat losses occurred in transit. This condition is not often approached in practice to a degree that will make these losses altogether negligible. It is nearly always advisable, to take into consideration the heat emitted by the pipe system, and to neutralize its disturbing effect on the differential pressure by such increases of volume as will re-establish the assumed uniform drop for which the system must be balanced. Perfect balance involves an extra amount of water passing each branch corresponding to the heat losses and thereby *securing even temperatures at return junctions, just as they are necessarily even at the junctions of the flow.* The total tax in B. t. u., including the losses, should then equal $W(t-t_1)$ with both factors constant, that is with $t-t_1$, and w_1-w maintaining between ends, but decreasing for the height of columns in question. The total difference in weight, or the actual pressure which governs the flow, is therefore reduced to an extent depending naturally upon the levels at which the heat emission takes place.

These variations from the assumed difference of temperature can be calculated, and the actual difference in weight between the two opposing columns established from the mean density in each section. This gives the pressure available in pounds per square foot or the head in feet of water of the density $\frac{w+w_1}{2}$. The differential height corresponding to the density w_1-w, on which the calculations may be based, is found by simple division of that total weight or height by the differential density assumed. The result gives the mean height of the total heat emitted on the circuit which must balance the resistance head.

The correct way of determining this mean effective height is the most accurate, but would require too much time in practice. A simpler method is to calculate directly from the heat loss and level of each sectional run, adding it to that of the appliance itself. A loss of 20 per cent. along flow and return lines, for instance, at a mean level of, say, one-half the height h of the radiator above the boiler, would give an effective head of $.2 \times .5h = .1h$. The radiator itself emitting the remaining 80 per cent. of head would produce .8h, making the true effective height available $.1h + .8h = .9h$, on which the resistance should be based. Expressed in a different form we find again the mean level

$$H = \frac{.2\text{B.t.u.} \times .5h + .8\text{B.t.u.} \times h}{\text{B.t.u.}} = .9h$$

which means that in the present case, the heat loss in transit, being below the point of delivery, will reduce its head by one-tenth.

Since the heat units emitted by the appliance and by each section of flow and return piping is already marked in the schedule for the purpose of determining quantities, the most convenient way to estimate the mean effective height will be to multiply each item of heat loss in B.t.u. by the height above boiler at which it takes place, and divide the sum of the products by the sum of B.t.u. The examples to follow will further illustrate the method.

It will be seen at once, that risers used as heating surface may have a decided effect on circulation, even when full allowance is made for the extra heat emission, unless they are sized for the corrected head. When balanced in this way, the difference $t-t_1$ will equalize itself at junctions, and the mean temperature will be even for each individual set of branches, but not necessarily for the heating appliance itself. With the greater part of the losses along the flow pipes, for instance, the radiator will be cooler, and *vice versa* with the losses greater on the return. Such deviations are appreciable only in odd cases, as occur for example with single main systems. They are properly corrected by increase or decrease of surface, making up for changes in efficiency in order to obtain the desired heat.

Deviations of the true effective head from the actual height of appliances, on the other hand, occur in nearly every apparatus, but are largely conditioned by the general scheme of distribution. The principal methods of piping should be considered mainly from this point of view.

The Method of Piping.—Gravity circulation, as ordinarily understood, can be effected by underfeed, overhead, and by single main distribution with secondary circuits. The same principles as expressed in a general way naturally apply to any of these methods, subject only to the assumption of the proper factors as to height and temperature. These must be studied and clearly defined for each individual case before the final calculation is attempted. It is assumed, of course, that an apparatus be designed to exclude disturbance by lack of escape for the air, or by the generation of steam.

Underfeed Distribution.—This method is distinguished by the location of both distributing and collecting pipes below the level

of heating appliances. The mains and branches usually run parallel, either in the same or in opposite direction. The general idea is illustrated by Fig. 9.

The heat losses along the mains, when insulated, are usually a small fraction of the total transmitted. While their bearing on the effective heights may become appreciable, it will affect all branches, more or less. The distribution is therefore not likely to be disturbed to any extent and no allowance need be made for the heat emitted by mains under ordinary conditions. For the branches, rising lines and individual connections, the losses are always relatively greater, and have more bearing on the head, especially when exposed, or only furred in, but not insulated. They should always be added to the tax in B. t. u. for flow and return, as if a part of the radiation. If these losses occur on runs serving more than one appliance, as for instance on risers leading to several stories, they should be apportioned in round figures to the tax on individual connections, since any extra volume carried in these main risers will necessarily flow also through the branches beyond, distributing itself among the latter along the path of least resistance. If, on the other hand, parts of the mains are to serve as heating surface, their heat emission must be added as an extra tax, distributed as mentioned above for risers, and its bearing on the effective height should be established for all individual circuits beyond.

When sufficient heat is generated so as to keep the flow temperature slightly higher, to make up for losses along mains, and the radiating units actually emit the desired amount, the assumed drop of 30° F., 35° F., or 40° F., should then maintain very closely between junctions, throughout the system.

As a general rule for underfeed distribution, it will suffice to base the losses of head and by friction and obstruction simply on the heating tax in B. t. u. rounded up to include the probable heat losses in branches, risers and connections, but bearing in mind the slight reduction of effective height due to the latter. The reductions of the effective height due to exposed rising lines and portions of mains, however, should be estimated from the heat emission and levels by one of the simpler methods, and taken into account.

Example.—Fig. 11 represents the pipe schedule of an underfeed system for a residence, calculated for temperatures of 180° F. and 150° F. in flow and return. The horizontal runs of flow

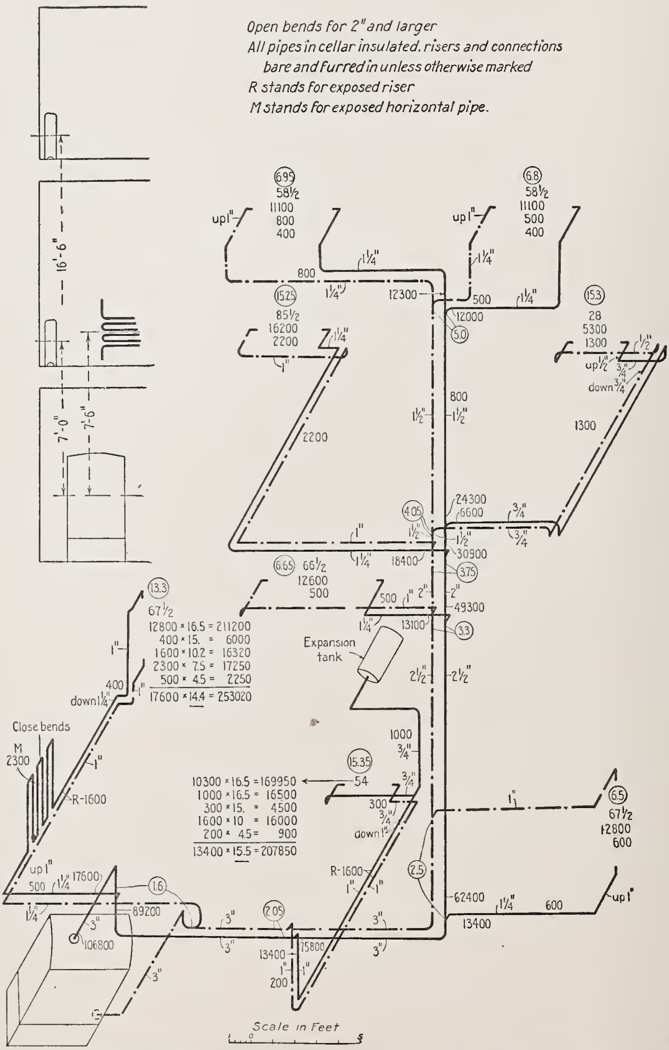


FIG. 11.—Example of hot water heating with underfeed distribution 180° F. to 150° F

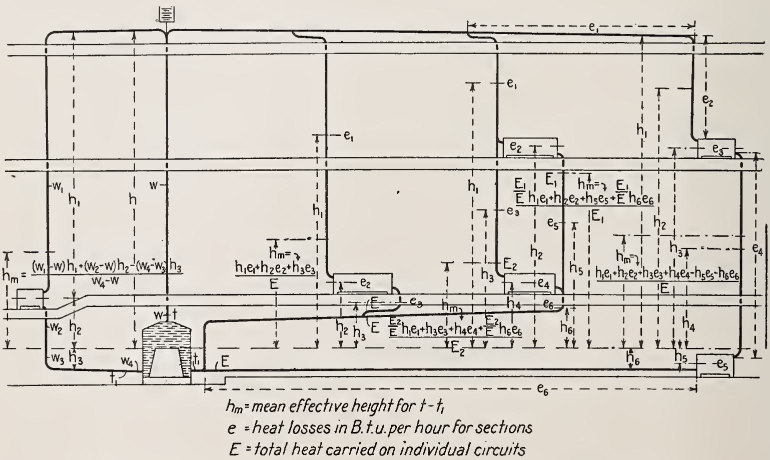
pipes were traced from the building plans and the vertical pipes are drawn to the same scale, under a fixed angle, so that all lengths appear in full and can readily be measured off. The levels of heating surfaces relative to the boiler are noted separately. The amount of surface to each radiator is stated, with the expected heat emission, depending upon style and height. The data in regard to insulation bearing on the heat losses in transit are also put on record.

The thermal units to be carried by each section of the piping are added up from the individual radiating units down to the boiler, including the losses for all risers, approximately determined and distributed among the individual connections. The preliminary sizes were then established on hand of the approximation chart. The final sizes are obtained by the graphic method of calculating and balancing resistances, the example giving the result of the final revision, carried as far as the best practice will require. The figures in circles indicate the actual height in feet which will produce the differential weight necessary to overcome the total loss of head in flow and return system up to the respective junctions. The first number (1.6) for instance, includes the resistance of the boiler, with the connecting flow and return mains, up to the first branch. The second figure (2.05) includes the first, with the losses for the following section of the mains to the next set of junctions added to it. The factors for branch tees are always entered at the beginning of a sectional run, for the continuation of the main as well as for that branch, on the basis of their respective sizes. Proceeding thus to the individual radiators, where the circuit closes, the sum total for each point of delivery indicates the approximation between the available and required heads. If the actual height is greater, there is head to spare, and the delivery will be greater. It will be noted, that the figures for the first floor are all below, but close to the actual difference of level, which is 7 ft. For the second story they come well within the actual height, but the depression of the effective height through heat loss in risers, and the disadvantageous position under throttling, make a greater factor of safety advisable for radiators on upper floors.

The example shows some instances where the risers figure as heating surface, giving the corrections therefore, as estimated from the heat emission at different levels above the boiler. In the case of a coil in a flow riser the depression of the mean ef-

fective height due to the chilling of that line on a lower level, is seen to be a considerable item. It would "spoil the draught," so to speak, for the radiator on that riser, unless the pipe sizes are based on this reduced effective height.

Overhead Distribution.—The distinguishing feature of the overhead system is a main flow pipe rising from the boiler to the highest point of the apparatus, with the distributing lines starting at that level and feeding the radiators from above. The returns are located below the level of radiation. A typical arrangement is illustrated by Fig. 12.



Fig, 12.—Schedule of hot water heating system with overhead main.

The main riser in this case represents the lighter ascending column, while the rest of the system, made up of the distributing and collecting mains, branches and risers, represents the heavier descending columns. The latter are naturally of the same total height, but in sections of varying density, the upper ones being made up of the feed lines to radiators, which are always a little cooler than the main rising directly from the boiler, and would create a circulation by themselves. With this arrangement the heat losses are, as a rule, relatively small in the main riser and greater in the branching feed lines overhead, thus placing the heat emission in transit at a higher level as compared with the underfeed, and thereby creating an increase of effective head or additional pressure. Whether this greater motive force is sufficient to overcome the extra resistance presented by the

main riser, will appear from the calculation of a system, with proper application of all factors. Unless structural or other reasons are decisive, this calculation should determine the method of piping to be used. Where radiators must be placed near to or even below the boiler level, the overhead system may give the opportunity to create sufficient extra head to assure circulation. In such cases it is often the only practicable solution.

Under ordinary conditions the heat losses in transit should be taken into account to the same extent as for an underfeed system. That is, they may be neglected for insulated mains, at least as far as they will affect all branches alike. Inasmuch as the differential weight is increased in this case, the branches further away are under no disadvantage. The heat losses on the individual branches and vertical feed and return lines from the attic to the cellar should always be added to the tax on the system in B. t. u. considering them as portions of the heating surface. The extra volume provided thereby will then practically make up the volume of water necessary to maintain the desired temperature difference between the flow and return mains.

If the overhead system is used with the idea of securing the increased mean effective head, the main riser should always be insulated, since any heat loss on the same will depress the mean level of cooling surfaces and reduce the head available for circulation. When the overhead feed is installed for different reasons, the heat loss can be made up by higher initial temperature, and does not bear on the distribution, as long as allowance is made for it. In either event, nearly all the heat emission with this method of piping occurs in the descending line. For appliances at low level, and at long distance from the heater the losses take place in greater part above the heating surfaces proper, and have relatively more bearing on account of higher mean temperatures involved. The risers for radiators at high level, near the main riser, on the contrary, show most of the losses below. The bearing of these facts on the head available should be established, not only for the proper balance of the system, but also for utilizing the opportunities peculiar to overhead feed, of creating extra head where most needed.

The method of doing this has already been outlined, and is further explained, as applied to a variety of conditions, in Fig. 12. With a single radiator on the circuit returning directly to the boiler, the problem is simply one of finding the sum of differ-

ential weights for the several parts of the circuit $(w_1-w)h_1+(w_2-w)h_2-(w_4-w_3)h_3$ and dividing the sum by the difference in density for the full temperature range (w_4-w) intended. This will give the mean effective height h_m for which the resistances are to be balanced. It is the accurate way of calculating the pressure available, as it takes into account the greater variation in densities at higher mean temperatures. For practical purposes, the process is too slow. The quicker and more convenient way is to figure again the mean level of cooling surfaces, from the heat emission of the individual runs e_1 e_2 e_3, which are already given, and their height above the boiler, h_1 h_2 h_3, which can be measured. With two or more units on the same set of risers as shown next, the heat emission on the common runs should be apportioned to the volumes passing the individual branches. In practice it will suffice to do this in round numbers, or to consider it simply in rounding up the figures for each branch.

The calculation of the effective head becomes of vital importance when parts or all of the appliance proper is located below the level of the heater, and the balance of the circuit must provide the head necessary for circulation. The same will apply when radiation is inserted in sequence, instead of in series. In such cases the mean temperatures for each unit should be estimated and allowance made for varying efficiency. This is also advisable with long exposed flow connections.

Example.—In the example shown by Fig. 13 the corrections to the effective height average about 2.5 ft., while for the underfeed it would probably be about $-.5$ ft. The extra length of the main riser uses less than 2 ft. additional head. The gain might reduce the size of a few connections, but would not permit a reduction in the size of the mains. Overhead distribution in this instance justifies itself rather on account of the fact that heat emitted by the overhead piping serves a useful purpose, and incidentally permits the entire system to vent itself through the expansion tank, which is a desirable feature in many cases.

For one of the circuits the temperatures and densities are figured accurately from the amount of water circulated and the stated heat losses along the line, giving the actual difference in weight and the water column for 40° F. range. The effective height for the branch is appreciably higher than that which

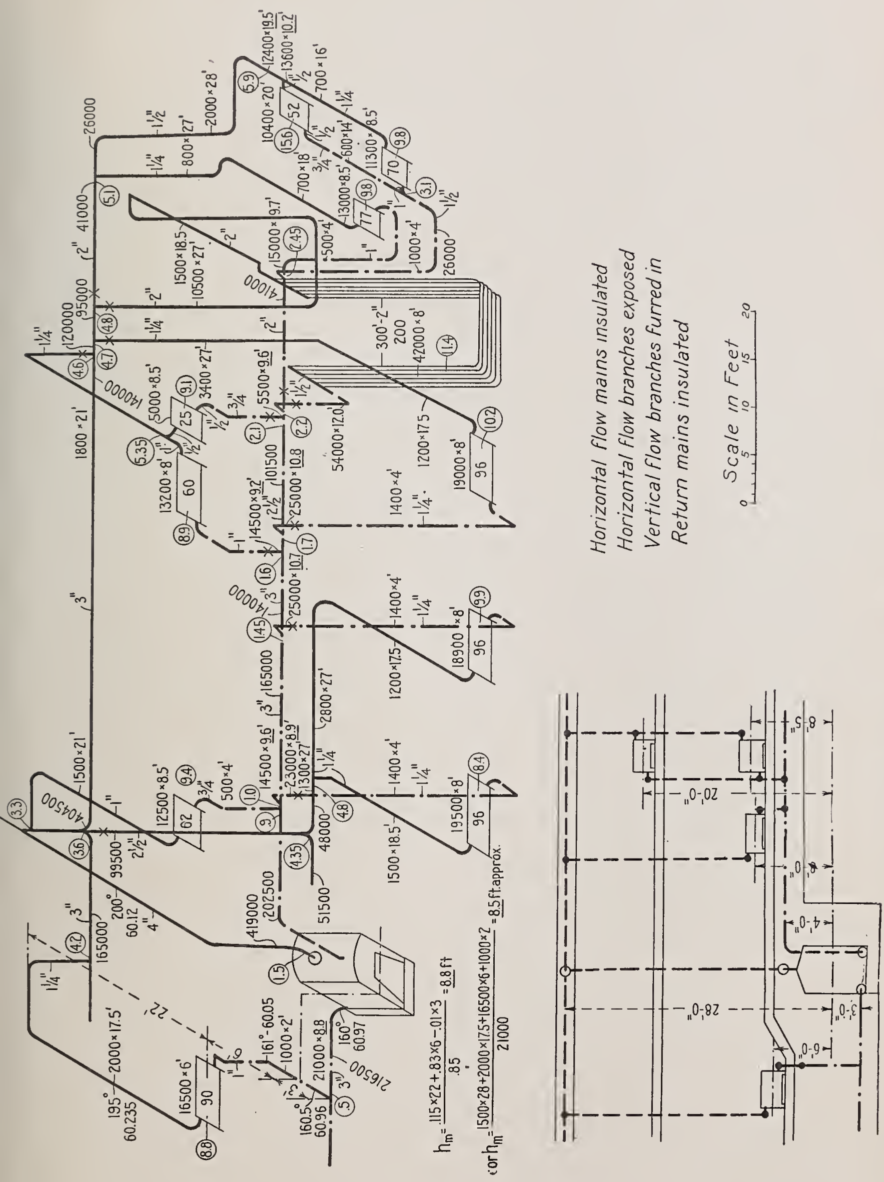


Fig. 13.—Example of hot water heating with overhead distribution 200° F. to 160° F.

was found by the easier method, but the difference is too slight to affect the pipe sizes in this case. For the other circuits the mean effective heights were figured in the usual way, as the ratio of $\frac{eh + e_1h_1 + etc.}{E}$, each item of loss being stated, together with the estimated mean level at which it takes place. The results are underscored and marked together with the sum E for each branch, at the respective junctions. The system is balanced for 40° F. difference, at 180° mean temperature. It will be seen, that the resistance heads (in circles) figured from the chart, are always kept a fraction below the mean effective head.

It may be noted in studying this example, that overhead distribution gives good opportunities for accurate balance by omitting or providing insulation for certain runs above or below the appliance with the idea of increasing or reducing the effective head for certain branches. Such expedients are not proper, of course, where heat is wasted thereby or becomes objectionable.

Single Main Distribution.—Fig. 14 is a schedule showing the method of piping generally known as the "one-pipe system,"

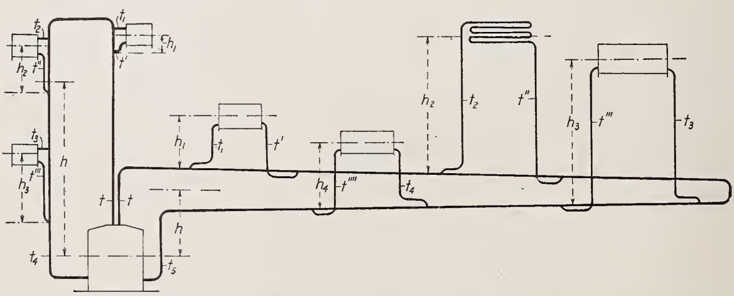


Fig. 14.—Schedule of hot water heating system with single distributing main.

but more correctly termed "single main system with secondary circulation." This single main, forming a complete loop between the outlet and the inlet of the boiler, may be considered as a distributing drum while the branches to the individual heat appliances are all separate, independent circuits starting from and returning to this drum.

When the temperature is kept up along the loop some circulation is almost bound to occur in each of the individual circuits. Distribution of the heat by this arrangement does not depend

as much on distances of radiators from the boiler, as it does on their elevation above the main and the general decrease in temperature along the same. The individual circuits are independent and do not have to be balanced among themselves. Fair results can be assured with this method of distribution by the old, wasteful rule of "making sure" by having the pipes large enough all around, which, by the way, will often fail with the two-pipe idea. Accurate calculation is nevertheless desirable for reasonably close distribution, to reduce the bulk of water, and for economy.

The first condition to determine is the drop of temperature along the main, under which it will circulate, and which should be kept small if it is desirable to maintain a high efficiency of heating surface throughout. For a greater drop the difference t_1-t', or t_2-t'' between the risers to and from appliances, should be larger at the warmer end, and smaller at the cooler end of the main line. Smaller differences result in higher mean temperatures, which will make up, to an extent, for the decreasing flow temperatures in the secondary circuits. The differential weight w_1-w thus changes for each set of risers and likewise the available pressure $(w_1-w)h_1$. The several sets should be figured independently for that reason, the problem being to find the size of pipe which will give about the right value for t_1-t', or that which would assure the expected efficiency of heating surface. Such graduation is not always necessary, where the loop serves a single room, or in other cases where the decreasing efficiency happens to be immaterial. If the main is looped vertically, as shown to the left on Fig. 14, the heights h_1, h_2, etc., can be figured out for a desired drop in temperature and the connections lengthened to make up these heights.

The effective height h governing the flow in the primary circuit is the difference between the mean level of the boiler and that of the several tappings where the cooled water rejoins the main. Since the difference in temperature should be small, the available height will generally determine whether the method is practicable or economical in a given case. A vertical loop usually permits a smaller drop to be maintained, and effects closer distribution, at less accurate balance. A horizontal loop is only advantageous when it can be placed at a good elevation above the boiler. It generally requires, the grading of individual branches

to make up for the decreasing temperature along the main. When well insulated, as it should be, the main is likely to emit only a small fraction of the total heat carried, but it will pay to take the losses into account, because, with a horizontal loop, they will increase the differential density for nearly the entire height available, and with a vertical loop usually for a considerable elevation. The heat losses for the risers and connections of the secondary circuits, especially when used as heating surface, should be included again in the tax, and allowance made for the reduction of the mean effective height in the same way as would be indicated for an underfeed or overhead system.

Example.—In the example presented by Fig. 15 the loop was first calculated for a difference of 20° F. between the two ends of the main, but the head required for a reasonable pipe diameter was found to be larger than the head room permitted. Since the elevation of the main above the boiler could not be increased beyond 8 ft. without prohibitive expense, the differential temperature at which the resistances would balance that height was found by tentative process to be 25° F. for a 6 in. main carrying nearly the total volume for the entire length. This range made it desirable to graduate the individual circuits, so as to keep the mean temperature of radiation within narrower limits. The resistances, accordingly, were decreased for the successive branches, starting at the flow end, so that the differential weight required, and the resulting temperature, would correspond about with the efficiency of surface as it decreases along the main circuit. This had to be done again by tentative calculation, using the factors for general corrections as to range, for a mean temperature averaging about 165° F. As will be noted, the drop for the successive branches decreases with the flow temperature toward the return end of the main, the lowest efficiency of the heating surface being thus kept up within about 85 per cent. instead of dropping to less than 70 per cent. without graduation. In the latter case, if no difference were made in sizes, the range would naturally increase toward the return end owing to the smaller differential weight at lower mean temperature, which must be made up by a greater drop. In this particular instance the drop would have increased about from 28 1/2° F. to 32° F. for mean temperatures ranging from 178° F. down to 153° F., as shown by the lines of equal height on the table of differentials.

To secure a fair graduation of the drop it has been necessary

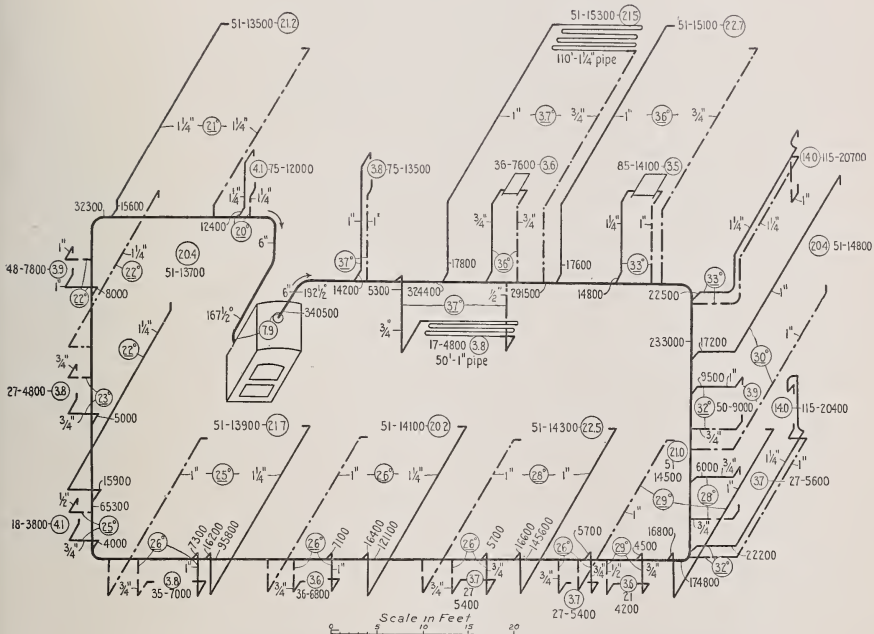


FIG. 15.—Example of hot water heating with single distributing main.

LEVELS AND DIFFERENCE OF TEMPERATURE.

Main pipe in cellar 8 ft. above mean level of boiler.

Drop in temperature between flow and return ends of main 25° F. Mean temperature 180° F.

From mean level of main to radiators on lower floor 4 ft.

From mean level of main to radiators on main floor 15 ft. (Effective height about 14.5 ft.)

From mean level of main to coils at gallery 23 ft. (Effective height about 21.5 ft.)

Drop in temperature for individual connections 40° F. to 20° F. Mean temperature 158° F. to 174° F.

Main pipe and branches insulated up to main floor.

in this case to use flow and return pipe of uneven sizes for some of the circuits. The necessity for this will occur more frequently with this method, if calculated as it should be, since the resistance head is made up by a single set of branches, usually very short, and cannot be varied by changes in parts of the main. The connections with the main, however, may be changed sometimes in length, with a view to modifying the resistances. This has been done in the example shown, where the length of runs between main and vertical lines is graduated in a general way, giving greater resistance where extra pressure is available. This expedient will also frequently help to balance an underfeed or overhead system. Local conditions, of course, must always determine or suggest the most practical ways and means.

THE FLOW OF STEAM

CHAPTER V

THEORY OF THE FLOW

Properties of Steam.—The relations of pressure, volume and temperature of steam are presented graphically by an adaptation of Dr. R. Mollier's[1] Entropy diagrams, based on Callendar's formula. The results from the latter deviate slightly from older data in certain respects, but seem to be borne out by more recent research. No attempt has been made to incorporate all the refinements needed for design of turbines and other steam motors. Although the diagram is made up in compact form, for the sake of better illustration, the readings of pressure, volume, temperature and density should be accurate enough for purposes of heating engineers. Separate scales are given for absolute pressures and temperatures. The latent heat can be deduced by subtraction from the total of the sensible heat, which is given by the temperature. The arrangement permits to follow the variations of the quality of the steam due to increase or decrease in the amount of heat, incidental to long distance transmission. It gives also the changes in pressure and volume, or work performed. The variation between the initial and final state can be traced by a line on the diagram, which will either approach the curves of equal heat or the lines of equal pressure. Rapid pressure drop for instance, due to high resistance, indicates the work performed and reconverted into heat, the line keeping more closely to those of equal heat. The extreme of pressure drop without heat losses is reached by the adiabatic expansion of steam, or the discharge through a frictionless nozzle, in which case the expansive force will bear on the volume delivered. In ordinary heating practice the changes follow more closely the isothermal lines, the variations of pressure being generally a small fraction of the total, absolute, under which the steam is transmitted. The analysis on hand of the table is useful in showing whether the changes in pressure and density as affecting

[1] Dr. R. Mollier's neue Tabellen und Diagramme für Wasserdampf. Springer 1906.

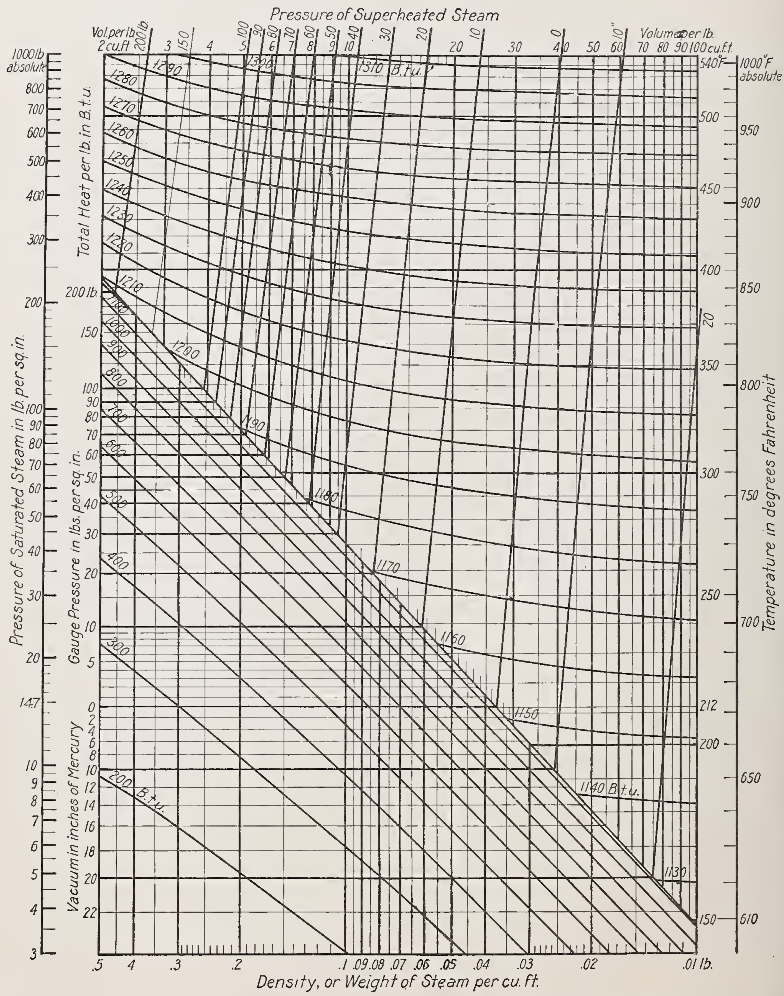


Diagram C.—Properties of steam. (Adapted from Dr. R. Mollier's Diagrams (1906), based on Callendar's formula.)

the volume and velocity are negligible in a given case. It will show also whether a certain pressure drop by resistance of a conduit, accompanied by a certain heat loss, will result at the end of the line in wet, saturated, or superheated steam. The lines will give the amount of condensation per pound of steam, or the degree of superheat, as the case may be. The variations in volume, due to condensation are also appreciable, and often bear decidedly on the resistance by friction and obstruction.

Friction in Pipes.—The velocity for calculating the resistances to the flow is naturally based on the volume passing a given run of conduit. It is expressed by $v = \frac{W \text{ per hour}}{3600\ wa}$ and must equal $\sqrt{2g\frac{P_v}{w}}$ wherein $P_v = P - (P_f + P_r)$, or the velocity head resulting under the friction and other resistances. W is the weight per hour, w should be understood to be the mean density maintaining along the run, and $\frac{W}{w}$ the mean volume. Since the density varies considerably for the range of pressure that may come in question, the quantities are better stated in units of weight, as the absolute measure.

High pressure steam for power purposes is usually measured in pounds per hour, or in h. p. based on the standard of 34.5 lb. of water evaporated from 212° F., or of 30 lb. from 100° F., into steam at 70 lb. pressure per square inch. As a heat carrier, it can be assumed to transmit roughly 1000 B. t. u. per pound, since the total energy given up when utilized for heating is nearly always more than the latent heat at 70 lb. The high-pressure chart is accordingly based on weight and approximate heating capacity, the horse power being derived by simple division with the equivalent 33,000 B. t. u.

Low-pressure steam for heating can be rated closely by the thermal units it will yield in condensation. If the pressure in appliances varies but slightly from that in conduits, the heat carried will not exceed materially that of evaporation. The quantities in B. t. u. from which the lines are computed therefore correspond to the latent heat per pound.

Among the formulæ giving the loss of pressure by friction in pipes, that of D'Arcy is often applied to the flow of steam. Expressed by the height of a column of steam it makes the friction

head $h_f = C \frac{Q^2 l}{d^5}$, wherein C is a coefficient varying with the diameter of the conduit.

Another generally accepted expression is one credited to Babcock for the weight of steam $W = C \sqrt{\frac{wp_f d^5}{l\left(1+\frac{3.6}{d}\right)}}$ or for pressure loss $p_f = \frac{W^2 l\left(1+\frac{3.6}{d}\right)}{C^2 wd^5}$. In this formula C is a constant. Babcock states it to be 87. Carpenter finds it to be 87.45.

Die "Hütte" quotes the formula derived from the experiments by Gutermuth, making the friction loss $p_f = C \frac{lW^2}{wd^5}$ which corresponds to D'Arcy's form for h_f, except that no variation of C is given.

As applied to high-pressure steam flowing through insulated piping under comparatively small drop, and with W as the initial weight, these formulæ have been found to agree substantially in practice. For low-pressure steam, where the loss of weight in transit is a greater item, as a rule, the results from formulæ by German authorities on heating, differ materially from the above. In so far as they cover the effect of condensation, the latter are likely to come nearer to the facts. The well known expressions by Rietschel, Fischer and others agree closely among themselves and may be accepted as the most accurate for the range of diameter and velocities met in heating work.

Fischer's expression of which all others are modifications, gives the pressure loss $p_f = \frac{al}{d^5}\left(\frac{W+0.5W_1}{2P_m}\right)^2$, wherein W is the weight of steam delivered, W_1 the weight of condensation dropped in transit. p_m the mean absolute pressure, and a is a constant applying to any diameter and velocity.

Taking the general formula used by Weisbach for expressing friction loss $p_f = w.f \frac{l}{d} \frac{v^2}{2g}$ and substituting for v its value $\frac{W}{3600\,wa}$ we obtain $p_f = w.f \frac{l}{d} \frac{W^2 4^2}{3600^2 w^2 d^4 \pi^2 2g} = Cf \frac{l}{d^5} \frac{W^2}{w}$ in which form C is a constant, and f is presumably variable, as for water and for

air. Comparing the results from these formulæ with those for water, it appears that the influence of diameter and velocity on the coefficient f is less pronounced with steam, or that the friction head varies more nearly as the square of the velocity, and inversely as the diameter. To find the approximate variation, the friction heads figured according to the above authorities were charted on logarithmic paper, and the average slope established for the lines of velocity and diameter. The angle of these lines gives the exponents for the hydraulic radius and slope in Tutton's general expression $v = CR^x S^y$ or $p_f = w.f \dfrac{v^{\frac{1}{y}}}{d^{\frac{x}{y}}} l.$

In this formula, v is the mean velocity as it may result under condensation and pressure drop for the length l, that is, v should correspond to the volume of steam passing a certain point somewhere between the two ends, representing the center of heat and pressure losses.

The average of lines for high-pressure were found by this graphic presentation to take on a distinctly different slope for those of low pressure, as derived from the German authorities, if based on an equal volume. The variation of the slope, or of the exponents for v and d would therefore seem to indicate the bearing of the density and temperature on the viscosity. This is confirmed again by the investigations of Biel previously referred to, which cover various kinds of fluids and conduits.

The value for f, derived from the charted pressure losses, with the exponents established, was found to be nearly equal to that for water. Inasmuch as the same kind of piping is generally used, the identical value has been assumed. Inserting these figures in the basic formula, the pressure loss by friction is for

high-pressure $p_f = w\ .0257 \dfrac{v^{1.95}}{d^{1.20}}$

for low-pressure $p_f = w\ .0257 \dfrac{v^{1.97}}{d^{1.16}}$

The exponents given here apply only to saturated steam at or near 70 lb. per square inch and for low pressure steam slightly above atmospheric pressure as used for heating. Superheating bears distinctly on the friction head, but these variations have not yet been fully established. Since the friction seems to decrease with the degree of superheat, it will be safe for most

purposes to assume the loss of head equal to that for saturated steam at the same pressure. The charts must necessarily represent average conditions. Inasmuch as they agree in general with the results from the standard formulæ quoted, they should give conservative values, at least for all conditions occurring in heating practice.

Local Resistances.—The general formula for the loss of pressure through various forms of obstruction as applied to the flow of steam, will show the exponents for v conforming to those given for the friction head. Thus we find,

for high-pressure $p_r = w\ 1.12r\ \frac{v^{1.95}}{2g}$

for low-pressure $p_r = w\ 1.07r\ \frac{v^{1.97}}{2g}$

The factor r corresponds again to the velocity head $\frac{v^2}{2g}$ at 10 ft., hence the constants are $\frac{10^2}{10^{1.95}} = 1.12$ and $\frac{10^2}{10^{1.97}} = 1.07$, respectively. The values of p_r for steam will decrease very little with higher velocities, so that for speeds above 10 ft. per second, they will be but slightly less than $r\frac{v^2}{2g}$. This shortage, however, is justified again, since the true coefficients for the high velocities in question are probably smaller than those generally quoted, which have been derived mostly from experiments at lower velocities, with water. The factors r for the various forms are identical to those for water in so far as the same style of pipework and fittings are used. The composite given for radiators includes the resistance of one angle valve only and an allowance for obstruction within, which is applicable to header coils as well. The return bend type should be calculated as runs of piping, at decreasing velocity, taken to be one-half the initial, assuming that no steam escapes at the return end. The factor for boilers includes the obstruction presented by outlets with nipples, header or dry pipe, in either of which the velocity head is lost and recreated repeatedly.

Probably because in common practice the pressure in steam heating systems is kept much higher than needed to effect distribution, the resistances to the flow have so far not received due attention. As shown later, it will pay to ease the flow of steam,

at least with certain methods of piping. Those types of fittings intended to reduce resistance are for that reason included in the list.

Velocity Head.—The losses of momentum as they occur in high pressure steam piping and heating apparatus are practically all incidental to the forms of obstruction as tabulated, the factors for which include them. Only a sudden decrease of velocity due to other features is to be taken into account. The loss of motion incidental to the drop in dynamic head is to be estimated and entered as a fraction of $P_v = w \frac{v^2}{2g}$, the values of which in pounds per square foot or squáre inch are readily found by the special line for the velocity head on all the charts. Gradual changes in speed, through tapers, seldom occur in heating work. The resistance of such special pieces may be estimated generally on the same principles as outlined for water and air, except in cases where expansion is liable to take place and affect the result.

The dynamic head at junctions and the relations to the static head often bear on distribution and are to be considered from similar points of view as explained relative to water. Occasionally the momentum is to be considered also when moist steam is carried and it is desired to separate the water.

Energy Expended in Flow.—Analogous to the work expended in pumping water, a certain energy is used in conveying steam through a system of piping, as expressed by the drop in potential caused by the movement. This energy, being reconverted into heat, offsets to a greater or less extent the losses by convection and radiation in transit. It affects the total thermal loss only in so far as the steam temperature is influenced thereby, and through the use of smaller pipe sizes resulting under greater drop of pressure. These thermal changes should be established wherever it is desirable to know the quality of the steam delivered, that is, its state of dryness as affecting the efficiency of motors, or when advantage is to be taken of the heat developed by friction and other resistances with the idea of assuring delivery of dry steam, or for economy in installation. The diagram of properties will be helpful in such investigations.

Expressed in B.t.u. per hour, the reconverted energy is $\frac{2545 \text{ W } 144 (P_f + P_r)}{\text{w } 3600.550} = .185 \frac{W}{w} (P_f + P_r)$. The equivalent for the work represented by the motion itself, or the velocity of dis-

charge which is $.185 \frac{W}{w} p_v$, can be considered as heat delivered. In practice p_v is usually a small part of the total pressure loss, and it is safe to assume the energy to be simply $.185 \frac{W}{w} P$, or $.185\, Q.P.$ Subtracting it from the heat losses in transit by convection and radiation, we find the net thermal loss in transmission, and from it the heat delivered. The relation of these two factors in reality determines the delivery and it would seem correct in principle to calculate a system of conduits directly on the basis of the thermal changes by the means of entropy diagrams arranged for the purpose. In heating practice, however, the energy expended in the flow is relatively small, and the drop in pressure alone will practically govern the discharge, hence it is made the basis for computation in the same way as for the flow of water and air.

CHART V

HIGH-PRESSURE STEAM

At 70 lb. per sq.in. Mean Pressure

THROUGH STANDARD WEIGHT IRON PIPES

Vel. in ft. per sec. $v=\frac{W \text{ per hour}}{3600\times w\times a}=\frac{\text{B.t.u. per hour}}{1000\times 3600\times w\times a}$ (at 70 lb. $w=.2$).

Pressure in lb. per sq.in. to overcome friction $p_f=\frac{w}{144}.0257\frac{v^{1.95}}{2g}\frac{l}{d^{1.2}}$.

" " " " resistance of obstruction $p_r=\frac{w}{144}1.12r\frac{v^{1.95}}{2g}$

" " " create velocity $p_v=\frac{w}{144}\frac{v^2}{2g}$.

Total pressure loss in lb. per sq.in. $P_1-P_2=P_f+P_r+p_v$.

Energy of flow in B.t.u. per hour $=\frac{2545W \text{ p.h.} 144(P_1-P_2)}{w\times 3600\times 550}=.185\frac{W}{w}(P_1-P_2)$.

The weight of steam or the B.t.u. transmitted should include the loss by condensation in transit on and beyond the run to be figured. To estimate the loss of heat, take roughly for

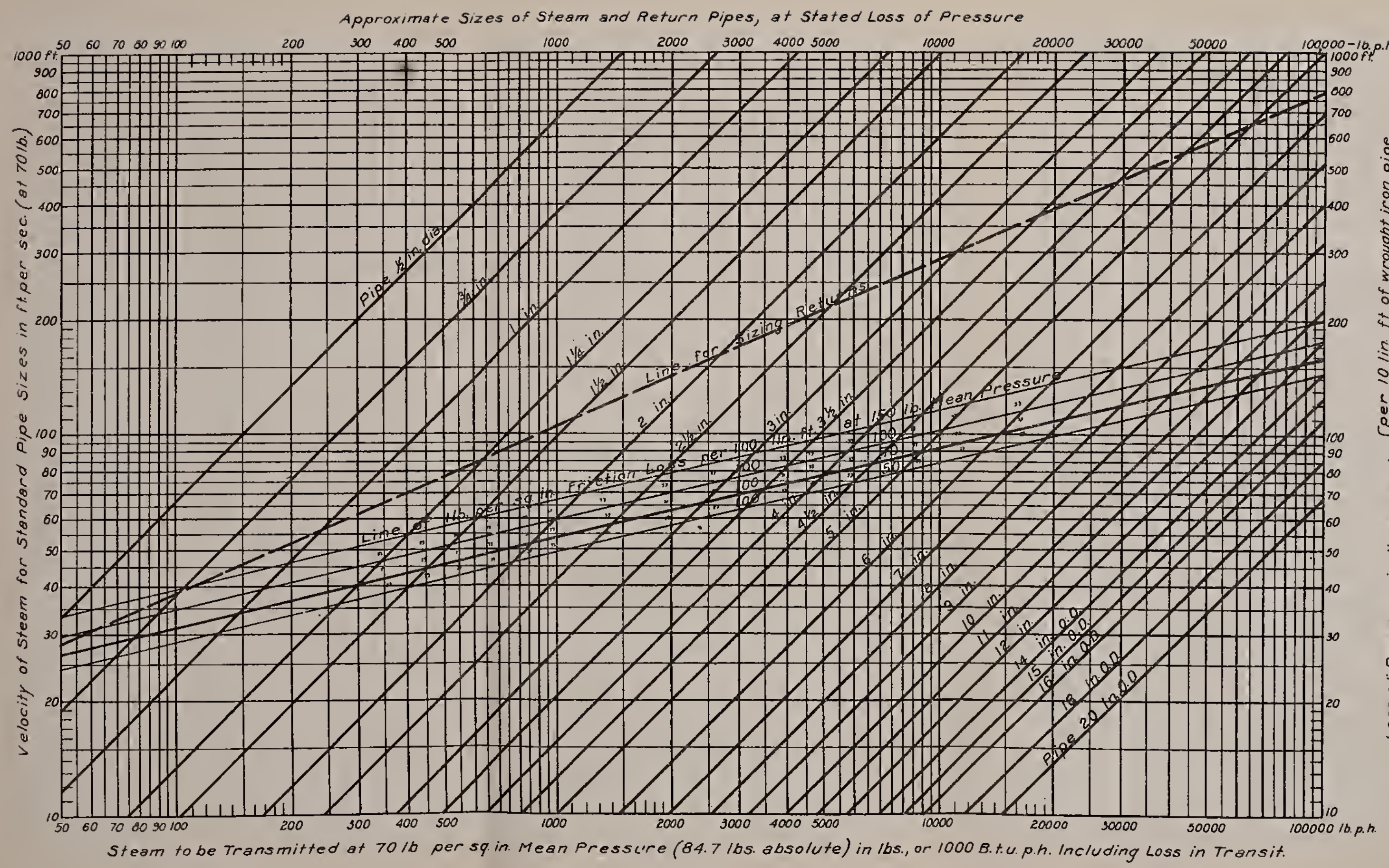
well insulated pipe, in conduits, per sq.ft. surface,	100 B.t.u. = .10 lb. p. h.
" " indoors, per sq.ft. surface	150 B.t.u. = .15 lb. p. h.
" " outdoors, per sq.ft. surface	200 B.t.u. = .20 lb. p. h.
bare pipes, in conduits, per sq.ft. surface	300 B.t.u. = .30 lb. p. h.
" indoors, per sq.ft. surface,	600 B.t.u. = .60 lb. p. h.
" outdoors, per sq.ft. surface,	900 B.t.u. = .90 lb. p. h.

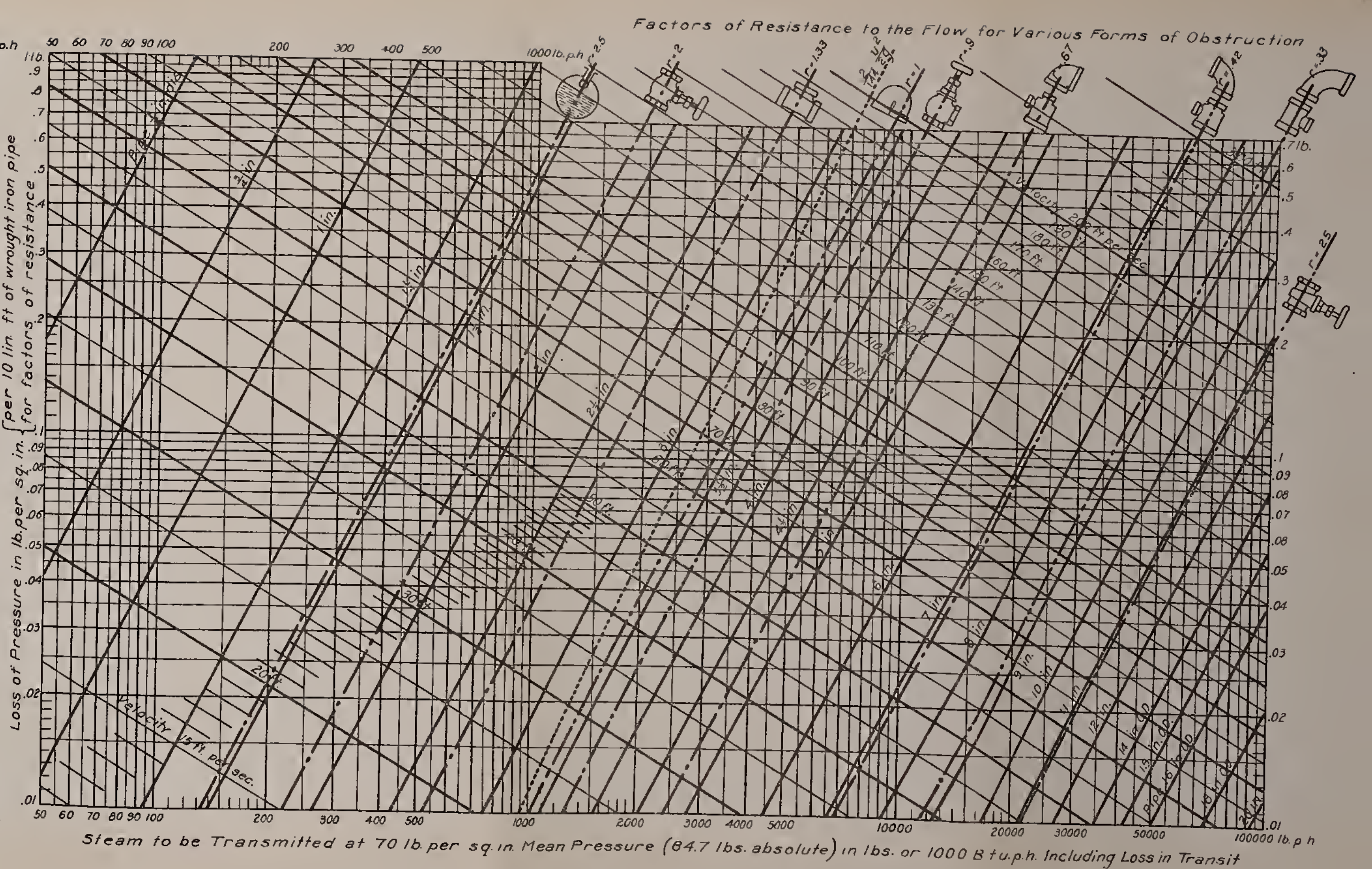
Corrections

When the volume decreases appreciably by condensation within a run of considerable length, or with rapid heat emission, it is necessary to figure straight long runs on the basis of mean velocity instead of the initial. Irregular runs should be figured in sections, each with the rate of flow through the respective part and any obstructions on the same.

For other mean pressures the losses by friction, resistance and flow, for the same weight transmitted, are to be multiplied by

3.06 for 10 lb. mean press.	1.00 for 70 lb. mean press.	.63 for 130 lb. mean press.
2.23 for 20 " "	.91 for 80 " "	.59 for 140 " "
1.76 for 30 " "	.83 for 90 " "	.55 for 150 " "
1.47 for 40 " "	.77 for 100 " "	.52 for 160 " "
1.27 for 50 " "	.72 for 110 " "	.46 for 180 " "
1.12 for 60 " "	.67 for 120 " "	.40 for 200 " "





CHAPTER VI

HIGH-PRESSURE STEAM DISTRIBUTION

Chart V.—The chart for calculating high-pressure steam piping is intended mainly for transmission of heat by live steam at long distances, where the pressure losses materially influence delivery. It is also convenient for the proportioning of pipes for power purposes.

The quantities are given in pounds of steam per hour, from which the B.t.u. carried for heating purposes can be figured by simple multiplication, 1 lb. corresponding practically to 1000 B. t. u. The pressure losses are expressed on this chart for a pipe length of 10 ft. in the customary measure of pounds pér square inch, as against pounds per square foot used for the low pressure diagrams.

The auxiliary diagram facilitates the approximate sizing of a line or system for different pressure to be carried, for a rate of loss of 1 lb. per 100 lin. ft. The line for sizes of returns will apply practically to either of these, except in cases where unusual resistance is to be expected, or when returns must carry a considerable amount of steam.

Data is added for estimating the heat losses in transit, which should always be looked into, and the quantities corrected accordingly. Factors are also given for correcting the losses by friction and local resistances for a liberal range of mean pressures, the charted values being accurate only for 70 lb. gauge.

Outline of the Problem.—The sizes of steam pipes should be determined by the drop of pressure permissible in transmission rather than by the absolute mean pressure carried. This principle applies to all classes of work. All problems in distribution and discharge should be considered from this point of view.

For power purposes, when the potential energy is to be utilized, the leading idea should be to reduce the resistances to the smallest practicable fraction of the initial pressure. When delivering steam to reciprocating engines, or other apparatus with intermittent flow, the pressure loss is difficult to compute. It will, in any event, seldom pay to attempt an accurate calculation since

the piping should be ample to act as an intermediate or supplementary steam reserve. Sometimes its weight and strength in resisting vibration are governing factors. But in any case it is desirable, as stated, to ease resistances and reduce the loss of potential as far as practicable. The chart for approximation gives fair average sizes for power work at various pressures, which may be rounded up or reduced according to length of runs and other conditions. Only for the transmission of power to distant points it is advisable to calculate the loss of pressure by means of the other chart in order to establish the initial pressure required, or to verify the final pressure available, as the case may be. No general rule of approximation can be given for such work. Each case should be considered on its individual merits.

For heating, the problem is essentially different. High-pressure is used either for the sake of high temperature, as for instance in drying coils, steam tables, cooking and a variety of technical purposes, or it may be used simply for the economical transmission of steam or heat at long distance. In either event the pressure loss, not the working pressure, is the factor governing the pipe sizes.

When steam is distributed to miscellaneous appliances, in use at odd periods, it will not pay to equalize the pressure drop in the pipe system for the quantities to be delivered at various points, but a rough calculation should be made of the total loss to the apparatus furthest from the source, or that requiring the highest final pressure. This will naturally lead to the sizing of the main line on the basis of the heaviest tax, and provide thereby a certain reserve capacity, which is always desirable under fluctuating load, if only on account of the extra flow for reheating. For the approximate sizing of such a pipe system the diagram giving the lines of 1 lb. per square inch friction loss per 100 lin. ft. will again be found convenient. The total drop of pressure is then to be computed by means of the other chart, with due allowances for mean pressure above or below 70 lb. per square inch.

Where high initial pressure is wanted for the economical transmisson of heat for long distances, the pressure drop may be a considerable fraction of the total, and equalization between several points of discharge is desirable, even though reducing valves are to be used. Theoretically, reducing valves are not

necessary, if the excess pressure is used up in transmission. To make the resistances of the line equal to this excess between the initial pressure and that desired at heating surfaces would be the ideal or scientific method wherever steam is delivered only at one point,or where the final pressure at several points is to be equal, and fluctuations in delivery take place at the same time. Whether the ideal method is practicable depends upon the conditions presented in individual cases. A long main line, for example, without branches up to a certain point, is advantageously sized to give such a pressure drop that will keep the steam dry, or will even superheat it by the friction, and to sacrifice, up to that point, the greater part of the available head. The further lengths of main and branches may then be equalized with sufficient accuracy by intelligent sizing, occasionally requiring a reduction for portions of individual runs. Branch pipes may figure very small when sized on this principle. There are, of course, practical limits to the velocities, particularly as to noise, although the probabilities in this respect are less than they will be if the steam is throttled at one point by a reducing valve which would present smaller area than a pipe reduced in size for a considerable length.

Distribution under Varying Pressure.—The condensation of steam in heating apparatus generally fluctuates within wide ranges, and the total pressure drop is about proportional to the square of the volume delivered. It seems essential, therefore, to investigate the bearing on distribution of variations in the delivery, which is controlled by the condensing capacity of the apparatus or appliance, if no steam escapes beyond. For reheating, the amount of steam is limited as a rule by the supply available, but it may easily be 50 per cent. in excess of the flow under normal conditions. When a building is overheated, condensation may be reduced often as much as one-fourth, even with all radiation in use. These variations are liable to be simultaneous for a whole system. Taking for example two branches, one to carry normally 1000 lb. of steam for a distance of 300 ft., the other 4000 lb. for 150 ft., each with a drop of 4 lb. at, say, 70 lb. mean pressure. On the chart for friction head per 10 lin. ft. the nearest size for the former branch is found for a rate of loss equal to $\frac{4\times 10}{300}=.133$ lb. per sq. in., which, for 1000 lb., is closely met by a 2 in. pipe. The 4000 lb. must pass through the other

branch at a rate of $\frac{4 \times 10}{150} = .267$ lb. and require a 3 in. pipe. The pressure losses figure 4.05 lb. and 3.9 lb. respectively. For a flow increased by 50 per cent., or 1500 lb. and 6000 lb. of steam, the corresponding losses are found to be 9 lb. and 8.7 lb., and for the decreased tax of 750 lb. and 3000 lb. we find the losses to be 2.30 lb. and 2.25 lb. The equalization under varying load, therefore, is affected only to a very slight extent, that is, only in so far as the velocity and diameter bear on the coefficient of friction. A reasonable number of local resistances would make little difference on this result. Since the resistances for steam, when compared with water vary more nearly with the square of velocity, the equalization may be said to maintain under the common fluctuations of load, such as are caused by the daily reheating and by weather conditions, when affecting all branches alike.

Distribution under Throttling.—The effect on the flow in one branch through the throttling of others depends largely upon the increase of condensation under greater pressure or higher temperature. Such increases of temperature within heating appliances as are likely to result through variations of the pressure loss in transit are appreciable in a distributing system designed for an excessive pressure drop. Taking the same example of 300 ft. of 2 in. pipe and 150 ft. of 3 in. pipe for illustration, and assuming they are supplied by a 3 1/2 in. main 600 ft. long, we find the pressure loss up to the junction to be about 12 lb. per square inch for the normal amount of 5000 lb. carried. With the 3 in. branch shut off, the 3 1/2 in. main will supply only the 2 in. branch, normally discharging 1000 lb. Leaving out of consideration the greater loss of heat in transit up to the junction, we find the drop of pressure in the main will be only about .57 lb. for 1050 lb. instead of 11.8 lb. for 5000 lb. through 600 ft. of 3 1/2 in. pipe, while that for the 300 ft. of 2 in. pipe is increased to 4.5 lb. for 1050 lb. The steam is thus delivered at a drop of 5.07 lb. instead of 15.8 lb., or at a final pressure nearly 11 lb. greater, with correspondingly higher temperature, in this case about 9° F., which would approximately increase the condensation to the extent assumed. Even in such an extreme case the delivery is therefore affected only by a small percentage and, ordinarily, throttling may be said to have little bearing on delivery in a closed system and with equal chances for steam to

condense. Conditions will differ, of course, when the steam is discharged through reducing valves, or delivered against atmospheric pressure. In the latter case the relative increase in the final pressure would control the discharge.

Periods of throttling or shutting of branches are naturally followed by periods of reheating, during which the flow to the respective branches is accelerated until the working pressure is re-established within the appliance. The excess of pressure on the main under throttling is then released, and will act as a reserve in meeting the extra tax, thus reducing the temporary disturbance in the flow to other branches.

All these effects, due to variation of load and to throttling, as they occur with high pressure steam distribution for heating, have bearing on the final pressure at the point of delivery rather than on the amount of steam actually discharged. But, as the final pressure would be greatest under light load, when least needed, it becomes necessary to regulate the initial pressure to meet the demand. For these reasons the ideal theoretical method of reducing the pressure altogether by friction in transit is rarely feasible in practice, and may be approached only for long runs of mains without branches. The reducing valve at the individual apparatus which marks the end of the line, would in reality have the function of a regulator, to take care of such fluctuations in the final pressure as are caused by the variant flow.

Application of Chart V.—To apply the chart, it is advisable to work up a schedule from the plan of piping on hand of which the quantity carried in all parts of the system can be figured and corrected for heat losses. Such a schedule should give all lengths to scale, indicate the features of obstruction and present, in general, all the factors entering the calculation.

As a rule it is necessary to use the chart for approximation on hand of the quantities without taking into account the losses in transit. For a working pressure of 120 lb. per square inch, for instance, the nearest size for 300 h. p. or 10,000 lb. of steam may be found between the intersection of the slanting lines of 100 and 150 lb. with those of weight or B. t. u. For long runs, or when the initial pressure is to be kept up, the next larger pipe, 5 in. should be assumed, while in the opposite case the next smaller size would be indicated.

The heat losses in transit should then be estimated on hand of the preliminary sizes, according to the class of insulation and

disposition, and the quantities corrected for all parts of a system before the pressure losses are determined. The latter may be obtained by the other part of the chart, starting at the end of the line and adding up the items for friction and obstruction from junction to junction for each run of even quantity and diameter. If the quantity decreases materially by condensation the mean value should be taken as a basis. According to the nature of the problem, the pipe sizes and pressure losses may have to be modified in order to effect equalization of pressure at all points of delivery, or to bring the total drop within a desired limit. The example below will further illustrate the method of procedure.

Example of High-pressure Steam Distribution.—The case illustrated by Fig. 16 is a central heating station in which high-

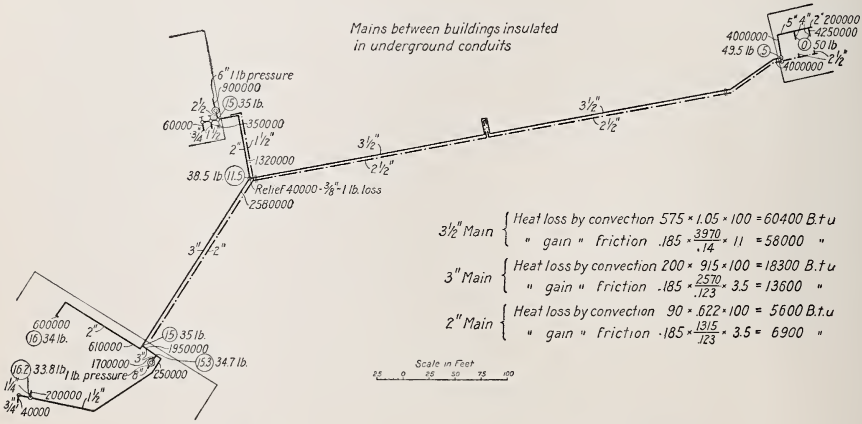


FIG. 16.—Example of high-pressure steam distribution for heating. Initial pressure 50 lb. per sq. in.

pressure is carried for a laundry near the boiler house, but which otherwise represents a simple problem of transmitting heat to various appliances and apparatus for a considerable distance. The working pressure at the boilers is 50 lb. per square inch, the highest final pressure required at the buildings is 10 lb. for certain fixtures. A drop of 20 lb. seemed not only permissible, but also desirable, in view of reducing pipe sizes and furnishing dry steam. It would still leave a fair margin for service at lower initial pressure, when the friction loss is

greater, also for reheating when less than 10 lb. final pressure will suffice.

A 3 1/2 in. main closely approximates these conditions. The heat developed by friction is nearly as large as the loss in transit, though it will not assure dry steam. This would only be obtained with a 3 in. pipe, which, however, reduces the pressure of delivery beyond a desirable margin. The pressure drop to the various branches is equalized as near as desirable. Reducing valves are used to permit a higher-pressure for various steam fixtures, and to graduate the pressure for heating the buildings according to the need.

Owing to the heat developed by friction, the quantities to be carried by the mains would not be materially increased through the losses in transit, except for the dry returns, the condensation in which must be added to the tax on the line. The relief, which connects the supply to a dry return main where shown, without a trap, is sized to pass sufficient steam to make up the condensation and equalize the system as far as possible. Theoretically, the pressure loss through this relief should equal that through the supply main, apparatus, and returns beyond that point. This will prevent short circuiting or back pressure in the returns, and at the same time avoid excessive suction through condensing effect, which might disturb the water level at the lower end of the line.

CHAPTER VII

LOW-PRESSURE STEAM DISTRIBUTION

Charts VI. and VII.—Of the two charts presented, that for 5 lb. pressure covers, in general, the field of heating by live steam at moderate pressure, while that for 1 lb. is intended for problems where distribution must be effected with the least drop in potential, at or near atmospheric pressure, that is, for heating by exhaust steam and for high class work generally.

The diagrams for approximate sizing simply chart the relations of quantity and velocity for the commercial sizes of standard weight iron piping, with slanting lines for different rates of pressure loss, for which a pipe system may be proportioned tentatively, or for purposes of estimating.

The quantities in both charts are stated in B. t. u. per hour, which is the generally accepted measure for estimating the heat transmission in buildings and emission or efficiency of heating surfaces. The B. t. u. as charted are based on the latent heat of the steam. The corresponding weight and volume can be found, if desired, on the table of properties. The true velocities may be read from either chart for any amount of heat and for all pipe sizes. The pressure losses are charted in pounds per square foot in place of the customary pounds per square inch since the various items of the calculation would be very small fractions of the latter unit. The pressure losses dealt with in problems of equalization have no direct relation to the working pressure carried. Hence there is no disadvantage in using another unit of measure. The total pressure differences from boiler to heating surfaces, for which it is recommended to equalize a system are stated as a guide for various conditions. The data for estimating heat losses are added, with a general statement concerning corrections and allowances.

Outline of the Problem.—Successful application of these charts depends again on familiarity with the factors entering into play, which can only be acquired by a study of the movements of the fluid under the widely different conditions presented by the various methods of piping.

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CHART VI

LOW-PRESSURE STEAM

At 5 lb. per sq.in. Mean Pressure

THROUGH STANDARD WEIGHT IRON PIPES

Vel. in ft. per sec. $v = \frac{\text{B.t.u. per hour}}{3600 \times 961 \times .05 \times a}$.

Pressure in lb. per sq.ft. to overcome friction $p_f = .05 \times .0257 \frac{v^{1.97}}{2g} \frac{l}{d^{1.16}}$.

" " " " resistance of obstruction $p_r = .05 \times 1.07 r \frac{v^{1.97}}{2g}$.

" " " create velocity $p_v = .05 \frac{v^2}{2g}$.

Total pressure for equalization $P = P_f + P_r + p_v$.

For two-pipe systems assume $P = 8$ to 16 lb. per sq.ft. according to distance.

" one-pipe systems assume $P = 4$ to 8 lb. per sq.ft. according to distance.

Static head in ins. water $H = .192 P_s$. $P_s = 4P$ allowing for reheating.

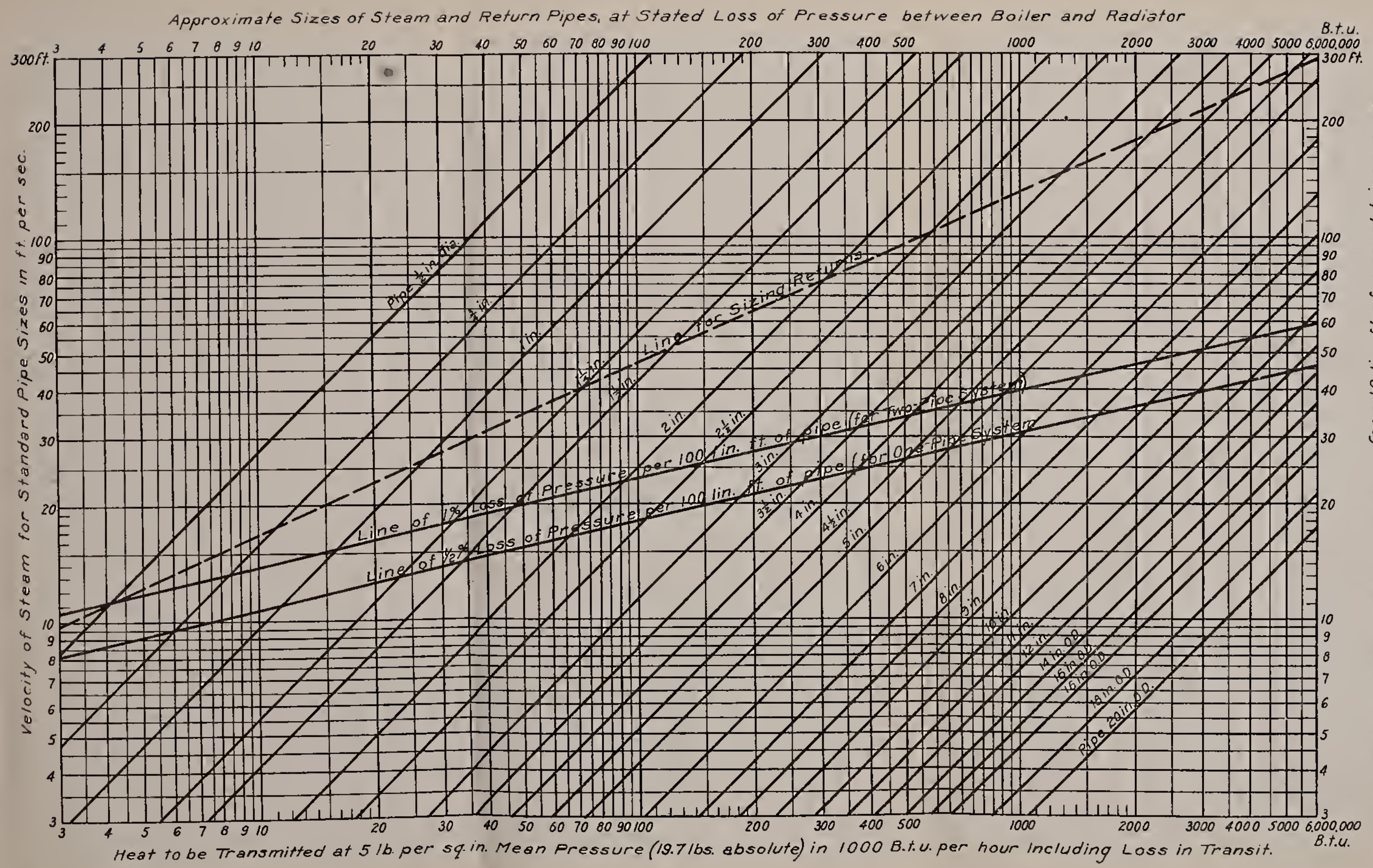
The B.t.u transmitted should include the heat lost in transit on and beyond the run to be figured, also that for any dry returns connected with it. To estimate these heat losses, take roughly for

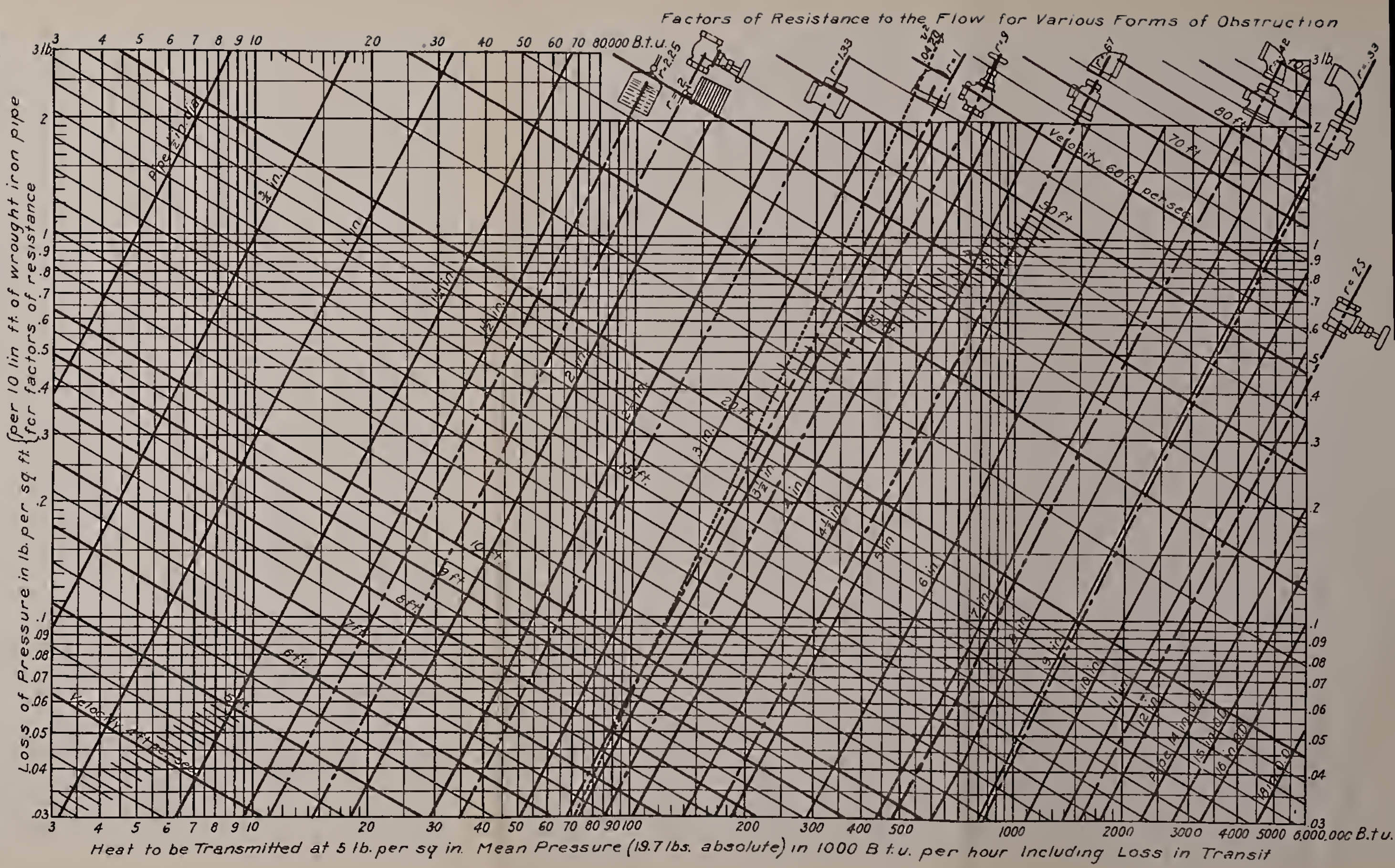
insulated pipes per sq.ft. of pipe surface,	110 B.t.u. per hour.
bare pipes furred in chases per sq.ft of pipe surface	225 B.t.u. per hour.
" vertical pipes in rooms per sq.ft. of pipe surface	350 B.t.u. per hour.
" horizontal pipes in rooms per sq.ft. of pipe surface	475 B.t.u. per hour.

Corrections

For exposed piping and long distances of even diameter the heat loss in transit should be included for one-half of the length only in order to obtain the correct value for v as mean velocity for the run to be figured.

More allowance should be made for static head under reheating when condensation in heating surfaces is accelerated by forced water or air circulation.





The flow of steam within an apparatus is governed partly by the condensing capacity of the several appliances, and partly by the carrying capacity of the pipes. These factors are interdependent to a greater or lesser extent. In a closed system, with all the air expelled, the pressure lost in distribution will bear on delivery only through the slight variations in steam temperature that may be involved. In any apparatus with fair circulation these temperature variations would affect the heat emission by a very small percentage only. With an open system, on the contrary, the volume delivered depends rather on the net pressure at the point of discharge, which may be the excess over the atmosphere, or whatever back pressure or vacuum is to be met. When operating under such low margin, the resistance of the pipe system will naturally bear on this final head available, which is the velocity head governing the delivery. Equalization of pressure losses is therefore essential to proper distribution in this class of apparatus. In practice, the closed or sealed system is never air tight or perfect in that sense. The actual situation is generally somewhere between the two possible extremes outlined, since the necessity for removing air, no matter how it may be done, will open any apparatus to an extent, either to the atmosphere, or to other influences affecting the net pressure and final discharge. To secure proper distribution it is always advisable, therefore, to assume the worst condition and to design the pipe system for equal pressure losses up to the ends of individual branches. How far it will pay to carry this equalization depends on the method of piping to be used, on the desirability of central or individual regulation of heat, and various other factors.

Central regulation, as far as it is feasible with steam heat, can only be affected by a general lowering or raising of the absolute static pressure within the heating surfaces. The pipe system and appliances of an apparatus that will permit this should be a unit, practically closed to the outer air so that any pressure may be carried, from that necessary for distribution up to the limit set for the maximum heating capacity. As previously demonstrated, distribution is not materially affected by variations in boiler pressure, and it is proper to base calculations on the maximum flow at the highest tension permissible, under a certain pressure loss. But in order to gain the full range of temperatures, distribution should take place at the smallest drop that

can be evenly maintained up to all points of delivery. If the means of expelling air assure fair conditions, the equalization need not be carried to a small fraction. But inasmuch as the losses increase about as the square of the volume of steam passing through the mains, which will often deviate considerably from the volume condensed within the appliances proper, according to the method of piping, it is necessary for even a fair degree of equalization to consider the system to be used. This should be done especially in reference to features which involve extra tax on mains and branches aside from condensation in transit. Dry returns, drips, reliefs and similar appendages or dead ends, for instance, will increase the volume to be carried, as also the variant means for removal of air and other special devices, which may entail a certain leakage.

Individual hand regulation by throttling devices, which give control over the admission of steam to the heating surfaces, requires a constant final pressure at the point of delivery. This must be closely equalized by friction or by special means of adjusting the pressure drop, as a vital condition to fair distribution. The returns of such a system are assumed to be open to the atmosphere, and the pressure carried within the supply piping and boiler is merely that necessary for distribution. An excess beyond that would cause a waste of steam, which would have to be checked by further adjustment or by traps. While in a closed system condensation takes place throughout, an open one theoretically passes no steam to the returns, and the total amount carried, and bearing on the resistance to the flow, is that condensed within radiation and supply pipes only.

The calculation of the distributing pipes will thus be modified according to the necessity for close equalization, and according to the excess volume to be moved which affects the loss of pressure and delivery. From these points of view the various methods of piping may be divided into three principal classes illustrated by Figs. 17, 18 and 19, representing respectively the wet return, the dry return and the open return idea. The bearing on the calculation of single or separate lines for steam and condensation, and of special devices for accelerating circulation, which may be applied with any or all of the main classes, are subordinate in importance from these view points, but should be considered in connection with each case. The effects of throttling, or the occasional shutting of branches, and of varia-

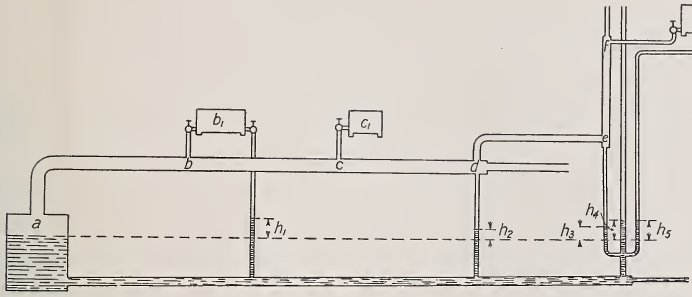


FIG. 17.—Steam distribution with wet returns, with or without vacuum or air lines, with single or double piping to individual radiators.

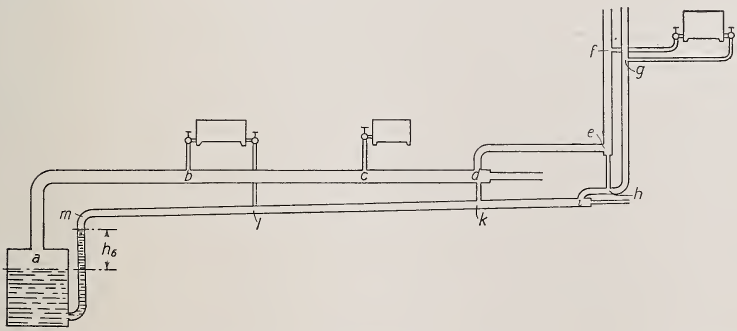


FIG. 18.—Steam distribution with dry returns, with or without vacuum or returns or air lines, with single or double piping to risers and radiators.

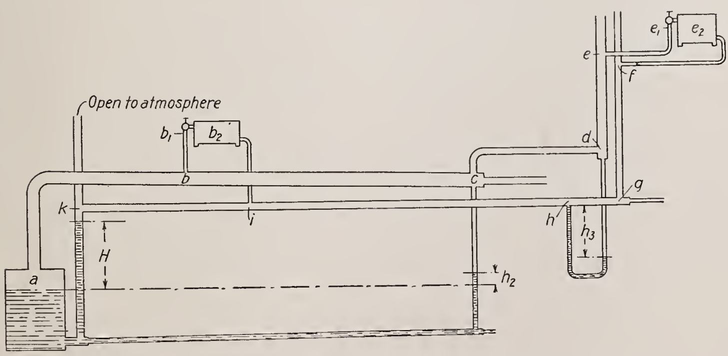


FIG. 19.—Steam distribution with open returns, with adjustable valves only.

tions in working pressure differ somewhat with the method of piping, though not radically. They bear on distribution in a similar way as pointed out for high-pressure steam.

Application of Charts VI. and VII.—The method of applying the charts is the same for any system of distribution and of regulation. It is always advisable to work out a schedule on the basis of the plans, in order to get a clear and comprehensive view of the situation in regard to the various points to be considered and in order to decide how far it will pay to carry equalization. The quantities and other data should be entered, the piping sized tentatively, and the heat losses up to and beyond the heat emitting appliances estimated and added according to the nature of the case, as will be outlined for each of the principal modes of piping. Allowances should be noted at the same time where condensation materially decreases the volume.

Figuring accurately, the quantity for any run of even diameter should include only one-half of its own condensation, but unless such losses are a considerable part of the whole volume passing that run, it is not necessary to split these individual items, as they are relatively small as a rule and the result is hardly affected whether the total is taken as a basis, or the mean quantity.

The final calculation of resistance heads may then be carried out with greater or less approximation, as the situation may demand. As a rule, it will be found convenient to establish and write down simply the losses, or differences of pressure from the boiler forward, from junction to junction, for all runs of even quantity and diameter. If equalization appears impracticable owing to high resistance between junctions, which would result in excessive differentiation between the branches at the near and far ends of the main, the size of the latter should be increased. In general, the ways and means to effect equalization are the same as indicated in the chapters for the flow of water.

For sizing returns, the charts give also a convenient way of proportioning, which may be modified, however, according to the method of piping used, and is explained in connection with the same.

The proportions of air lines are also touched upon in dealing with the various systems.

Wet Return System.—Fig. 17 illustrates a pipe system with a wet, or sealed return. Steam fills the entire system down to the

CHART VII

LOW-PRESSURE STEAM

At 1 lb. per sq.in. Mean Pressure

THROUGH STANDARD WEIGHT IRON PIPES

Vel. in ft. per sec. $v = \dfrac{\text{B.t.u. per hour}}{3600 \times 967 \times .04 \times a}$.

Pressure in lb. per sq.ft. to overcome friction $p_f = .04 \times .0257 \dfrac{v^{1.97}}{2g} \dfrac{l}{d^{1.16}}$.

" " " " resistance of obstruction $p_r = .04 \times 1.07 r \dfrac{v^{1.97}}{2g}$.

" " " create velocity $p_v = .04 \dfrac{v^2}{2g}$.

Total pressure for equalization $P = P_f + P_r + p_v$.

For two-pipe systems assume $P = 6$ to 12 lb. per sq.ft. according to distance.

" one-pipe systems assume $P = 4$ to 8 lb. per sq.ft. according to distance.

" open vapor systems assume $P = 4$ to 8 lb. per sq.ft. according to distance.

Static head in ins. water $H = .192 P_s$. $P_s = 4P$ allowing for reheating.

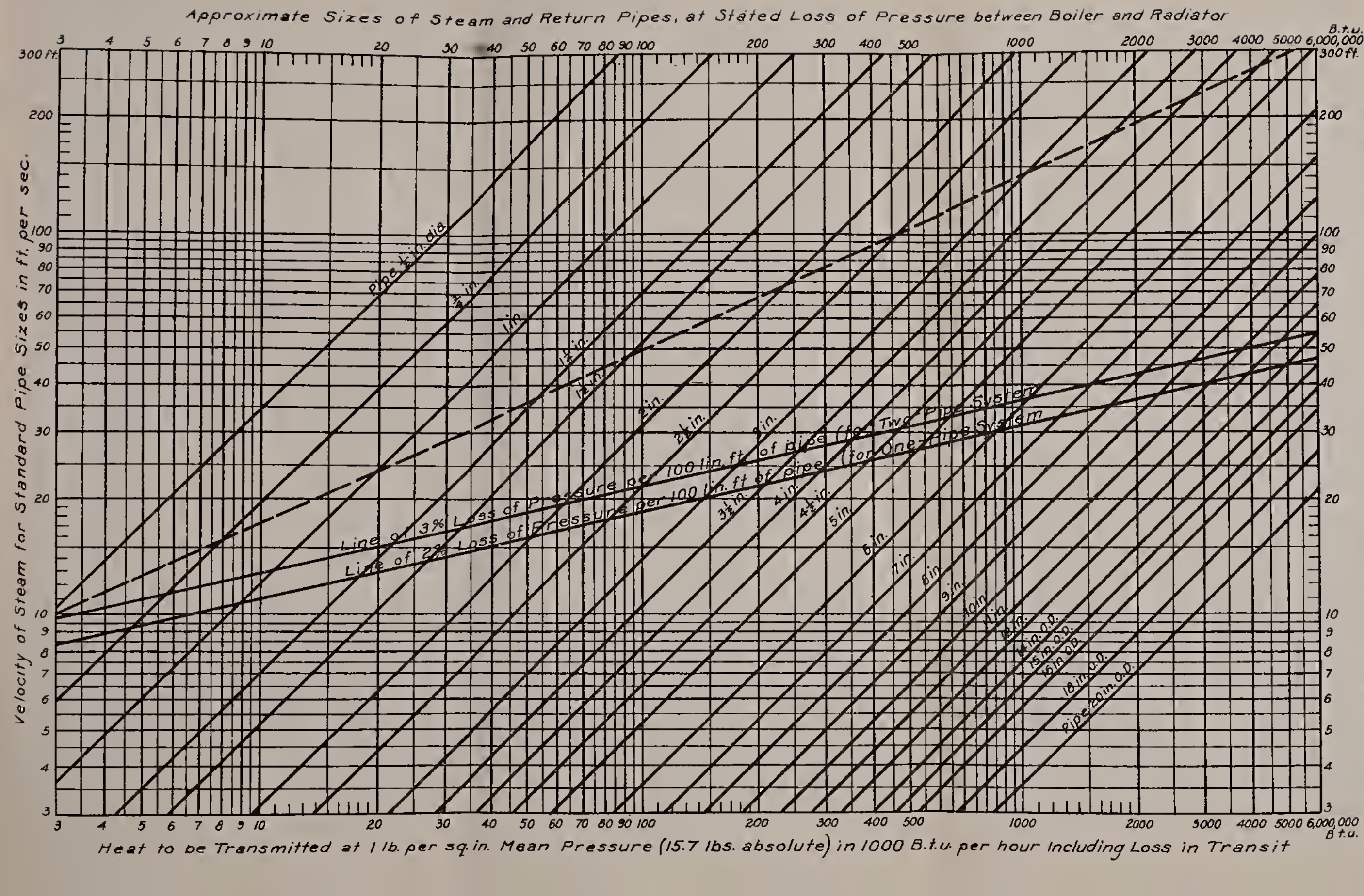
The B.t.u. transmitted should include the heat lost in transit on and beyond the run to be figured, also that for any dry returns connected with it. To estimate these heat losses, take roughly for

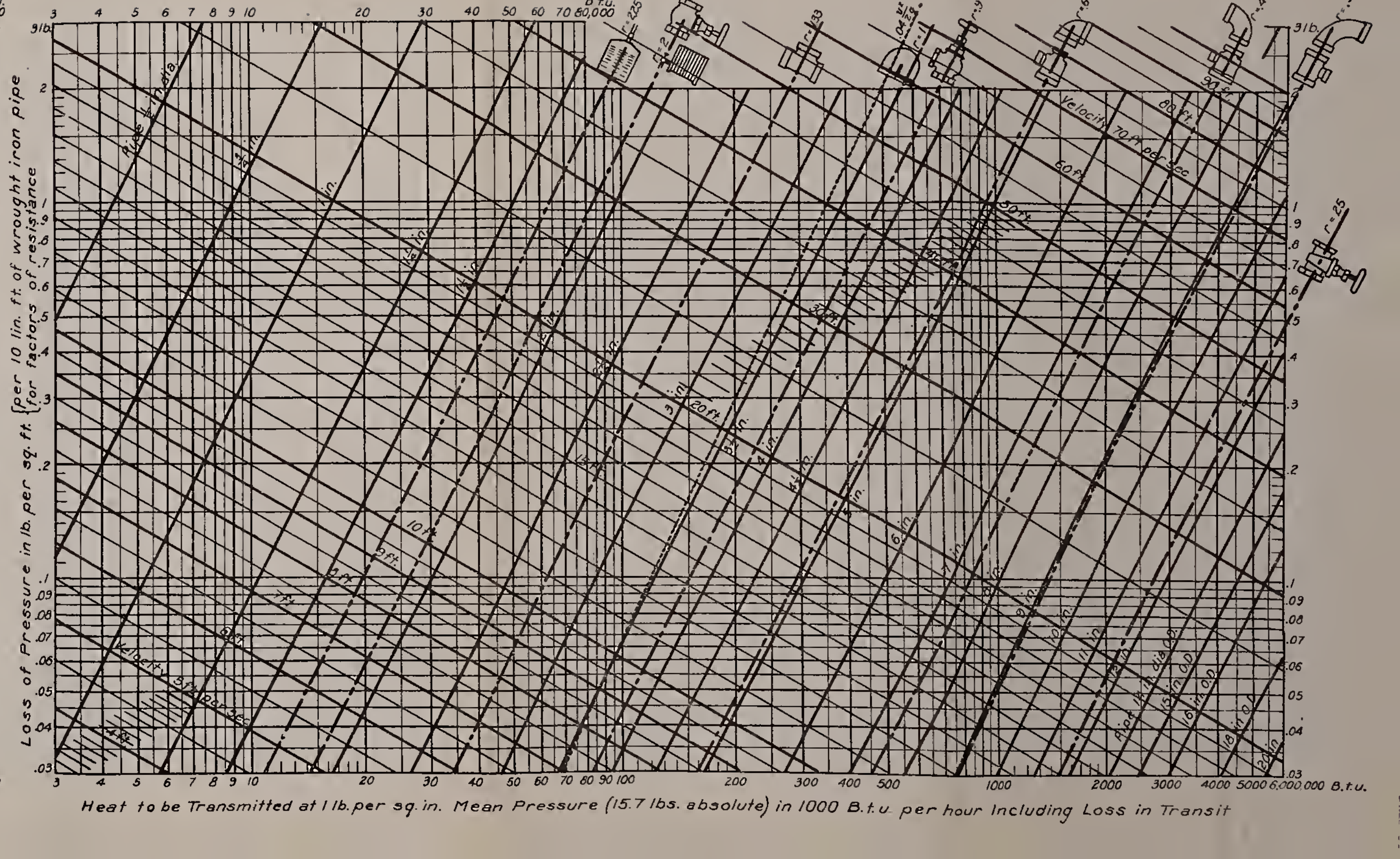
insulated pipes per sq.ft. of pipe surface,	100 B.t.u. per hour.
bare pipes furred in chases, per sq.ft. of pipe surface,	200 B.t.u. per hour.
" vertical pipes in rooms per sq.ft. of pipe surface,	300 B.t.u. per hour.
" horizontal pipes in rooms per sq.ft. of pipe surface,	400 B.t.u. per hour.

Corrections

For exposed piping and long distances of even diameter the heat loss in transit should be included for one-half of the length only in order to obtain the correct value for v as mean velocity for the run to be figured.

More allowance should be made for static head under reheating when condensation in heating surfaces is accelerated by forced water or air circulation.





water line. The bulk of it condenses within the appliances, but a considerable part is always dropped in transit and in the dry portions of returns and reliefs. The size of steam supply pipes at a given point is clearly to be based on the volume passing that point, or on the total condensation beyond it. The steam condensed in exposed return risers, for instance, should be included in the volume passing into a radiator, and added, with the losses in the supply pipes, to the amount of steam passing through the mains.

For a systematic calculation of a distributing system the pipe schedule should show the general situation in respect to heat losses ahead or beyond appliances, based on the preliminary sizes, as well as the length of runs and items of resistance to the flow. The aim should be to establish as near as possible the true quantities, or approximately the volumes, carried in each part of the system. The resistances for each run of even diameter and velocity, from the boiler up to the point of discharge may then be determined with sufficient assurance from the charts and equalized by selection of proper sizes.

As stated in a general way, the drop of pressure in the main itself should be kept down so that the individual connection furthest from the boiler will not have to be excessively large in order to come within the desired limit, while the first connection should not figure much less than the customary size in order to make up the same total pressure drop. If this total is equalized in this way, the static heads h_1, h_4 and h_5 or the water lines in the returns will be on a level. The height of these heads above the water line in the boiler will then correspond to the loss of pressure by friction and obstruction up to the appliances, provided, the return pipes offer no appreciable resistance. Likewise, the static heads h_2 and h_3 in the relief pipes indicate the pressure losses in the mains up to the points d and e. It is not necessary to equalize the static head in these sealed reliefs, since the water cannot rise higher in them than in the sealed returns from appliances, but it will pay to establish these heights where the steam mains are placed near the water line. For the same reasons the pressure drop permissible in a sealed return apparatus is limited sometimes by the level of heating surfaces. Mains and appliances should be located well above the calculated normal water line in the return since the pressure loss and the static head will be much greater under the extra tax on the pipes

when heating up. With the volume of steam temporarily doubled, for example, the water will rise to four times its regular height, hence a very liberal margin should be provided beyond the calculated head. A fair height for the mains above the water line in boilers is given by the simple formula $H = .192\ P_s$, taking $P_s = 4P$, or roughly four times the total pressure loss in pounds per square foot up to the relief or return pipe in question.

With single pipe lines carrying all their condensation against the steam it is advisable to allow for the extra resistance by doubling the losses found by chart for these runs, thus calling for larger sizes to obtain the same drop of pressure.

As a rule, close equalization is not essential for a sealed return system, but it is advisable to compute the pressure losses at least approximately, in order to render account of the differences in water level created, and, as previously pointed out, with the idea of better general regulation, as well as simultaneous reheating.

Theoretically, the method of removing air, whether by hand, automatic valves, or by suction in various forms, should make no difference on the flow of the steam in the supply piping, if the air is removed without allowing steam to escape. The air valve is presumed to be closed whenever the heating surface is in full action, hence there should be no extra volume of steam passing beyond, and no increased tax on the mains. The flow of steam and pressure drop are increased only when vacuum is turned on during reheating, or through leakage. If evenly distributed, the increased net pressure created may then be assumed to make up the extra pressure loss due to greater volume.

The free opening of ordinary automatic air vents bears on the pressure loss during the period of reheating, in so far as a rapid escape of air will accelerate the condensation, but the discharge through the valves depends again upon the pressure available for expelling the air, which may be affected by back pressure through the air lines. For heating systems working under very slight pressure difference, especially when exhaust steam is being utilized, under vacuum or not, it is desirable, therefore, to equalize, approximately, the back-pressure produced by a system of vents. It would be impracticable to estimate the volumes of air, and, in any event, to carry out a detailed calculation, but a fair equalization should be secured by maintaining a certain

proportion to the steam supply pipes. If, for instance, air lines are sized for one-fourth of the diameter of the corresponding steam supply pipes, they will generally prove to be ample, and bear out the best practice. It is recommended also to size the air valves accordingly and to use larger patterns for appliances calling for larger steam connections. This rule would tend also toward larger air lines and valves on long runs of mains, from which more air is to be expelled, since long runs will always call for larger sizes, if properly calculated.

In a wet return system, with each drip from the individual appliances sealed, very little steam is usually carried in the return piping above the water line, as it represents only the condensation in the dry portions. Whatever the amount may be, it will rarely enter as a factor in sizing the return pipes, the idea being to make them large enough to practically eliminate any additional loss of pressure on all branches beyond the point of delivery, at which the supply pipes are equalized. Since it would not pay to investigate in each case, or to calculate whether the friction head in the wet portions exceeds a certain limit, a general rule of proportioning may be figured out which will keep the resistance of the water at or below a certain ratio to that of the steam. This ratio would naturally maintain also for the period of reheating, when the amount of water increases at the same rate as the volume of steam and would, to a degree, equalize the losses in the returns or neutralize their effect.

Since the volume of the condensation is only about .0007 times that of the steam at 1 lb. pressure, the resistance head, in a wet return, for a uniform proportion of one-half of the diameter of the corresponding steam supply pipes figures only about 1/20 of that within the steam lines. This pressure loss in the return should be added to the total in the steam mains, but as it is nearly alike for all branches, it can hardly bear on distribution and may be neglected altogether for purposes of equalization.

For practical reasons, this proportion recommended is not applicable to the smallest branches. Hence the lines for sizing the returns on the approximation diagram will be found to graduate them so as to make the minimum size 1/2 in. This relative increase for the smaller piping brings the pressure losses down still further and makes it inappreciable, at least for pipes running full, without carrying steam.

If the sizes of the steam lines are materially changed by the

final calculation, it will be proper to correct the returns to conform to the rule of half diameter, but discretion may be used in such cases.

Example of a Wet Return System.—Fig. 20 is a working schedule of a typical low-pressure steam heating apparatus on the two-pipe plan, with sealed returns. The system includes direct and indirect heating surfaces emitting the maximum heat required at a pressure of 1 lb. per square inch, while distributing evenly at the least practicable excess above the atmosphere.

The tax on the mains is not greatly in excess of the total for the heating surfaces, except for a few exposed rising lines and their returns. Aside from the latter, the condensation in transit amounts to only 40,000 B. t. u. or less than 10 per cent. of the net total. With moderate sizes for the mains, corresponding practically to those given by the approximation diagram, it has been possible to equalize the losses in branches, risers and connections, without resorting to undesirably small or large connections, at a pressure drop of about 7 lb. per square foot, or only about .05 lb. per square inch. If the automatic air vents are in working order, this pipe system will circulate throughout before any pressure is recorded by an ordinary gauge, and could be operated as a so-called vapor system. The equalization might be made still closer by increases or reductions of sizes for parts of connections, but the effect on the heat distribution would hardly be appreciable. The ample pipe sizes resulting from the small pressure drop for distribution at normal operation provide a safe margin for extra tax and will permit rapid reheating without fear of disturbance. The total pressure loss under reheating $4\,P = 28$ lb. per square foot may in this case raise the water line in the extreme end of the system by $.192 \times 28 = 5.4$ in. There would be no danger, therefore, of any water backing up into the stacks, if the latter are placed about 6 in. above the water line in boiler. Check valves are not needed on the returns when the possible pressure losses are known and taken care of, nor is water likely to be entrained even with a flow considerably in excess of that for regular service, because of the moderate steam velocities necessary for equalization. In cases where such liability exists, due to peculiarities of boiler design or other causes, it would be advantageous to enlarge the main line up to the first turn or the first branch, thus reducing the initial velocity still further and the pressure drop throughout.

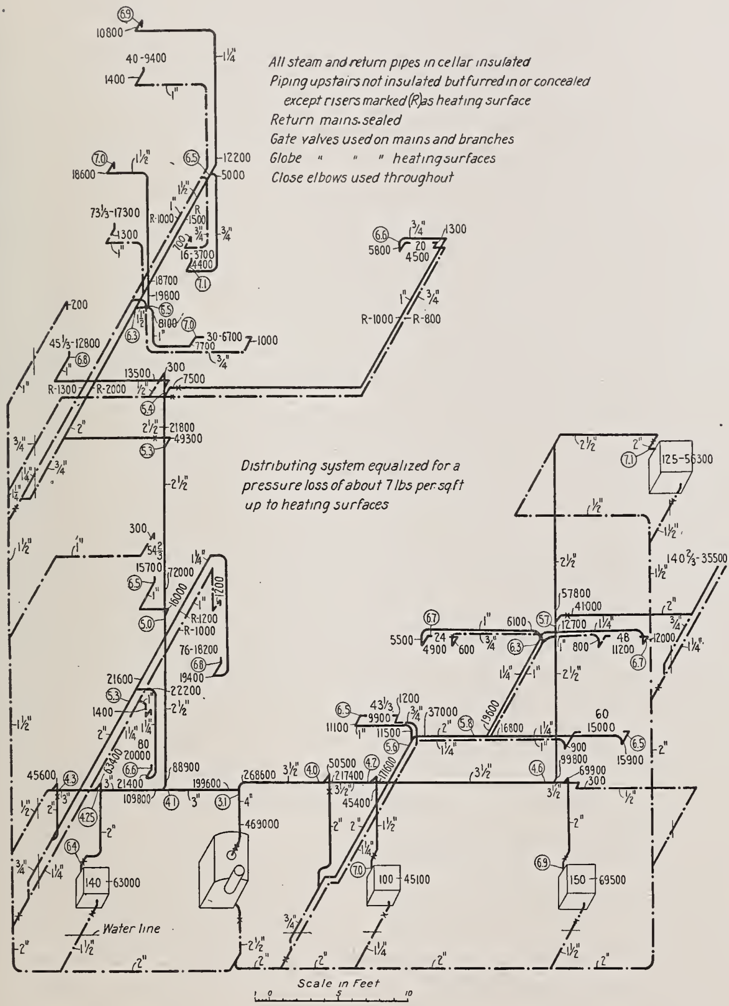


FIG. 20.—Example of low-pressure steam heating at 1 lb. pressure. Wet returns—two-pipe distribution.

Dry Return System.—The dry return idea, as generally understood, differs from the wet return only through the disposition of the water seals, which are placed on the mains, instead of being put on the individual branches. The essential feature to be considered is the possibility that steam may pass from the supply pipes into the returns and back up through appliances, thus giving opportunities for short-circuits which do not exist with a separate water seal for each individual branch. Strictly speaking, any wet return apparatus with double risers to several stories is a mixture of the two systems, unless the condensation from each radiator, on all stories, is carried in separate lines, sealed before entering the main. Return risers with several radiators should, in fact, be calculated as dry returns.

The equalization of such a system, in order to assure smooth, even circulation, should be carried out with the idea of avoiding short circuits. Referring again to Fig. 18, the pressure losses, theoretically, should be calculated to be equal for the runs a–b–l–m and a–b–c–d–k–l–m and a–b–c–d–e–f–g–h–i–k–l–m. In other words, whatever steam passes through branches and reliefs into the dry return mains at l, k, i, and h, should be no more than necessary to replace condensation in the same and move toward the boiler. An excess entering at l, and passing up through k, d or i, h would run against the condensation and may disturb drainage. The quantities of steam necessary to keep the return mains filled, especially the non-insulated portions, will thus become a factor, not only in sizing the relief pipes, which should pass that steam under a certain head, but also in the computation of the resistances for the supply piping. The heat emitted by these mains adds often materially to the tax on the pipe system. Bare return risers, for instance, should always be figured as heating surface, and the steam included in the volume passing into the nearest radiator above. Return risers should thus be filled through the radiator connections above them, while the horizontal runs are naturally taken care of in the same way through the reliefs.

Unless a dry return apparatus is proportioned on this principle, there is liable to be a counter flow of steam in some parts of the system. At the period of reheating this counter flow of steam is increased and is liable to meet the first condensation, which is always cold. When meeting cold water, this steam is condensed at a very rapid rate and increases the tax on the

mains still further at a critical time, when the system is already taxed beyond the normal. The disturbances often resulting from this cause have made dry return mains unpopular. When correctly sized, dry mains can be used safely without artificial devices for drainage.

The flow of steam within the dry returns, induced only by the condensation within the same, causes but a small part of the total pressure drop from a to m. Nearly all of the resistance is met in the runs up to the heating surfaces and up to the points k and h on the reliefs. It is not necessary, therefore, except under unusual conditions, to carry out the theoretical equalization down to the water line at m which indicates the total loss. The logical thing to do will be to equalize again up to the appliances only, but making full allowance for condensation in all dry returns and to reduce the losses in the latter to a point where they cannot become disturbing factors. Reliefs from main to main should be sized to present equal resistance as any other branch, for instance up to k and h, treating the portions of dry return mains to be filled as heating surfaces.

It has been found in practice, that dry returns should be larger than wet returns. Proper calculation, taking full account of all the losses in transit, will result in the first place in appreciably larger steam mains. If the returns are then sized on the same proportion of 1/2 diameter, they will also be larger, but it will often be desirable to allow for the steam to be carried with the water. To what extent this is advisable depends, of course, on the individual case. While larger pipes will generally help, they will also increase the condensation. It is always better to treat each case on its merits, by proper equalization, that is, by increasing the sizes beyond one-half of the diameter of the steam pipe, wherever the resistance might become appreciable.

Dry return mains may be used also with single rising lines, carrying steam and water. These should be computed in the same way, with reduced drop, as recommended in connection with a wet return main, except that the next relief should be proportioned so as to pass about the right amount of steam with the water.

The method of air removal will bear on the flow in a dry return system in a similar way, as with sealed returns, if arranged through air valves on the appliances, and *vice versa*, the final pressure in radiators will affect the action of vents. Air lines

may be proportioned accordingly on the same plan as recommended for the wet return system.

If vacuum is applied, it can be made to act in this case through air lines or through the returns. No increase of flow can be counted upon, if the vacuum is automatically controlled at the return ends of radiators. No decrease in pipe sizes seems justified, therefore, on the assumption that the vacuum will help the flow, if the air vents will shut properly, as they should. Under regular operation the flow will again be governed by the amount of steam condensed. Where it is desirable to reduce the working pressure below the atmospheric, a continuous leakage is necessary, and the return ends of radiators are to be left open to the suction. The extra volume of steam passing will then depend principally upon the surplus of steam available or to be condensed, and the capacity of the jet or other device for disposing of it, and creating the vacuum. Allowing for such a surplus, which puts an additional tax on the steam mains, and considering also that the volumes and friction losses are still further increased under lower mean pressure, it is best not to reduce sizes, even though a larger pressure drop can be secured under vacuum. When the vacuum must be created by motive power in one form or another it is naturally advantageous to keep down resistances. It seems proper, in any event, to calculate for the pressure that is likely to be carried in extreme weather, and effect equalization under a moderate drop between the boiler or reducing valve and radiators. The return mains likewise should not be reduced on account of the use of vacuum, inasmuch as the tax on them may be materially increased thereby, and an effective transmission and equalization of the vacuum itself is highly desirable.

Example of a Dry Return System.—The pipe schedule illustrated by Fig. 21 shows a low-pressure steam heating apparatus for a bank building that presented somewhat unusual conditions. Dry returns were necessary in some parts to avoid pipes on the floor or in trenches, and in other parts in order to utilize them as heating surface. The tempering coil returns are kept dry to avoid backing of the water from one section into another under extreme condensation. The individual return branches thus communicate, but the main branches unite below the water line and are separated. The total tax on the system is 1,611,000 B.t.u., while the heating surfaces proper call for only 1,359,600

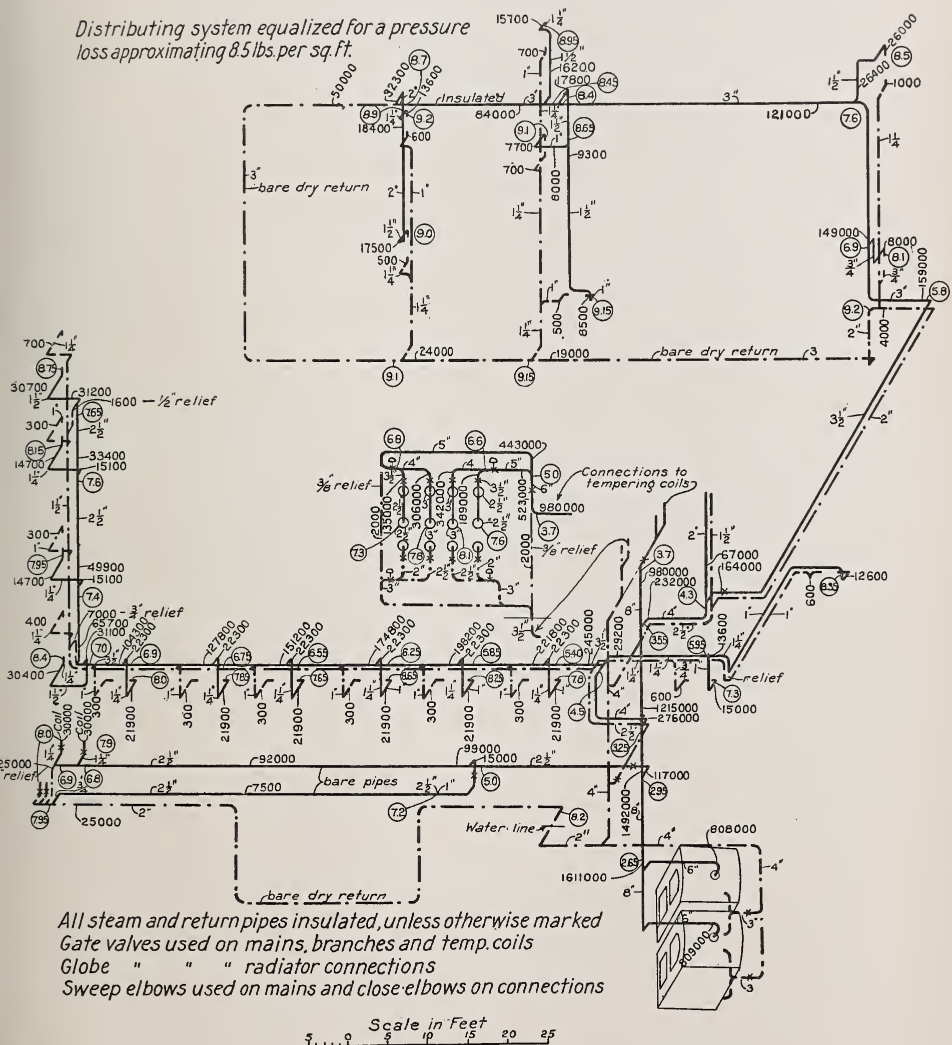


FIG. 21—Example of low-pressure steam heating at 1 lb pressure. Dry return—two-pipe distribution.

B.t.u. The condensation in transit figures therefore 251,400 B. t. u. or over 18 per cent. of the heat to be transmitted. Of this latter amount the dry returns, partly utilized, call for 94,300 B.t.u., causing a very decided extra resistance in some portions of the system. Even if all were insulated, they would be an appreciable factor in the calculation.

The mains figure somewhat larger than the nearest sizes from the approximation table. The equalization is not as close as in the preceding example, but keeping within 10 per cent. of the average pressure loss of 8.5 lb. per square foot. These discrepancies could only have been reduced by further increase of mains or by reductions of sizes in odd places. As it is, the pipe system assures a very fair distribution of the heat and responds quickly, without noise, while steam is being raised.

Although the calculation of pressure losses in the connections to tempering coils is necessarily problematical, owing to the fluctuations and uncertain condensing capacity of the several sections under automatic control, it will pay to carry it out, as near as practicable, to avoid excessive pressure differences between the several sections, which would cause disturbance in the returns. The ample sizes in which such a calculation invariably results, will also neturalize the effect on the other parts of the system of the sudden flow to coils under the operation of automatic regulators.

The reliefs are sized to pass sufficient steam to fill the sections of returns to the nearest relief, or to the water seal below, at a pressure drop making up as nearly as practicable the total for equalization. These reliefs are desirable especially where all radiation may be shut at times and cannot be depended upon to pass any steam to the return main. The resulting vacuum may, under certain conditions, disturb the water line.

Example of Low-pressure Steam Heating at 5 lb. Pressure.— The schedule shown by Fig. 22 represents a typical one-pipe system, with returns for the horizontal mains only. A part of the latter is dry, the other part is sealed. The total tax on the boiler is 203,000 B. t. u. while the heat emission of the radiators aggregates only 166,000 B. t. u. The balance of 37,000 B. t. u. is mainly heat from the risers which is generally utilized in this class of work, but not always duly taken into account when sizing the pipes. It represents, in this case, 22 1/2 per cent. of the net tax and increases the pressure loss in the mains by

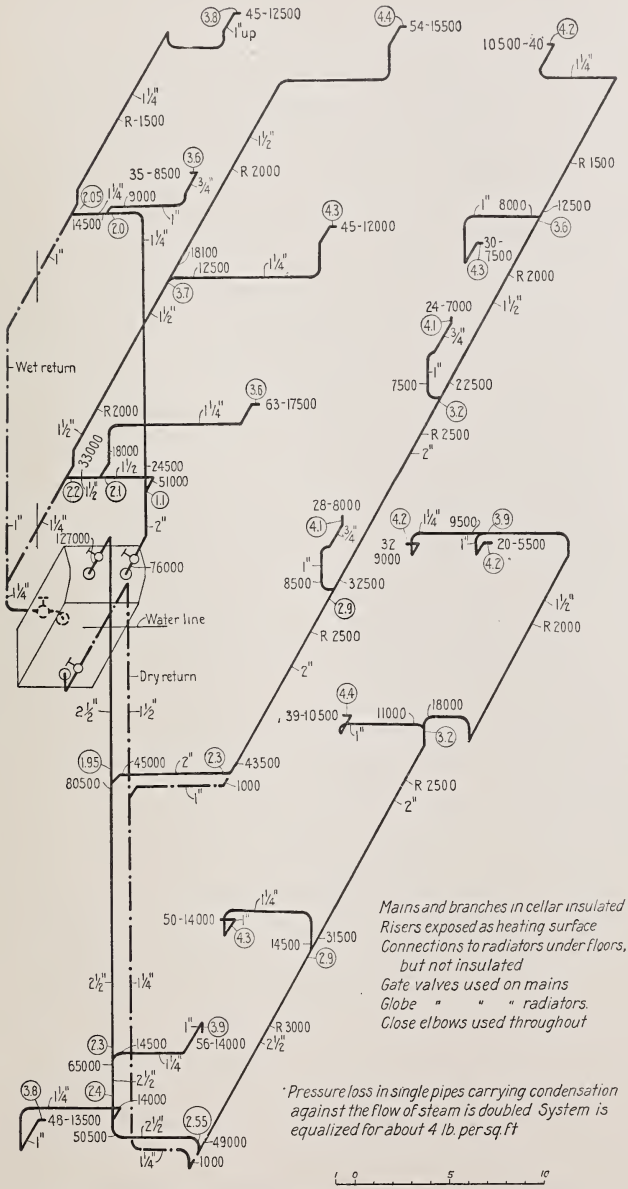


Fig. 22.—Example of low-pressure steam heating at 5 lb. pressure. One-pipe system.

nearly 50 per cent., thus becoming a decided factor in the calculation.

The equalization is carried out to within 10 per cent. of the stated total drop of 4 lb. per square foot without excessive sizes of mains, and without reducing the area of near branches beyond a point at which the condensation would not flow freely against the steam. A safe limit in this respect is given by the line of pressure losses for one-pipe systems on the approximation table, which should be considered as giving the smallest size advisable for single connection draining backward. If the size of the nearest branch, thus determined, makes the pressure drop much smaller than desired, it is best to modify the sizes of the main, with the idea of reducing the losses between junctions, or deliberately to take off the first branch at a point further away from the boiler, thereby increasing its length and total resistance. Each problem should be studied individually in order to find the best and most practical means to effect fair equalization.

With a sealed return main the circulation will always be more positive, or under better control. This will be apparent when it is attempted to figure the sizes of the reliefs from rising lines discharging into a dry return. Such relief pipes must pass the condensation from the riser and all its heating surfaces, also the steam that is needed to fill the return main down to the water seal. It is practically impossible to calculate the pressure drop in the relief under these conditions and it will be best to proportion them as returns, keeping down the sizes of the reliefs near the boiler in order to prevent a counterflow of steam to those further away. Thus the return and relief on the first riser on the dry end of the system in Fig. 22 is made only 1 in., while under usual conditions 1 1/4 in. would be the proper size. Owing to the greater tendency for the steam to rush through at this point, there will be no difficulty in draining by the 1 in. pipe provided.

A one-pipe distributing system computed on this plan can be made to circulate as well as any arrangement with double piping and other improvements, but if properly carried out, the saving in first expense will not be as great as is generally assumed, since the pipe sizes undoubtedly should be more liberal, even at higher boiler pressure, to allow for the extra resistance of the water flowing against steam.

The Open Return System.—Returns and air lines combined,

and open to atmospheric pressure, are only feasible when the discharge of steam through each branch is restricted to the amount which the appliance will condense. This restriction may be obtained either through the resistance to the flow presented by the boiler outlets, piping, valves and heating surfaces, or by special throttling devices permitting adjustment and graduated control of the volume for a stated initial pressure. The latter may be lowered for purposes of a general reduction, or central regulation of the heat, but must not be exceeded, lest steam be wasted through the open returns. Again, the full working pressure must not overbalance a water column limited to the height of the return main above the water level in the boiler, as indicated by H on Fig. 19. Higher initial pressure would seal the return, prevent the free escape of the air and destroy the possibility of graduating control over the steam at the appliances, which is the principal reason for using this method.

Reliefs from the steam mains are to be separated from the returns from heating surfaces, or they may be trapped to prevent discharge of steam. The static head h_2 in the relief corresponds to the pressure drop in the steam main, while the head h_3 in a trap is the difference between H and the loss of head up to point d. These seals must be carried low enough to allow for this.

When properly adjusted, the return system of such an apparatus will carry only water and air. The volume of steam entering into calculation is, therefore, strictly that condensed by the heating surfaces and the supply piping. The loss in transit, moreover, is liable to be smaller in a well designed open return system than in a closed one, since it is desirable to insulate the steam supply pipes thoroughly for better control over the flow.

Successful hand regulation by graduation valves depends upon close regulation of the working pressure and also upon close equalization of the pressure losses. Theoretically, the pressure drop should be calculated from the boiler up to the individual or common outlets to the atmosphere, usually a stand pipe at the end of the return main, making the total for circuit a–b–b_1–b_2–i–k equal to the total for a–b–c–d–e–e_1–e_2–f–g–h–i–k. The resistances in the combined return and air lines, or what might be called the back-pressure on the radiators, would be difficult to estimate. It is best to proportion the piping again to practically eliminate its effect on the flow by liberal and consistent

sizes of returns. The question will then arise whether the system should be equalized up to the controlling valves at points b_1 and e_1 or up to the return ends of appliances b_2 and e_2. In the former case, if an equal pressure is actually maintained behind each valve, the discharge through the same must still be adjusted to obtain the right resistance for the valve and the appliance itself. In the latter event, the resistance of the graduating valve and the radiator itself is included in the total pressure drop, and no adjustment would be necessary, in theory, if the factors of resistance for these valves and other parts of the distributing system were accurately known, and the plant carefully calculated and executed. The uncertain elements in the computation and construction make it necessary to depend on adjustments in either case, but that process is made easier and favors more accurate graduation if aided by previous calculation as far as possible and leaving the least amount of adjusting to be done by the fitter.

The factors of obstruction for the graduating valves differ greatly with the type, design and the full net opening, which is often much less than that of the nominal pipe size. According to Zyka's experiments[1] on the discharge capacity of such valves under various heads, the factors based on actual velocity at smallest area would figure about $r=2$, or equal to that for a straightway globe valve, or for an angle valve with radiator. This includes the velocity head lost in discharge. The thin jet of steam issuing from a valve of this type probably represents about all the loss of motion taking place within the radiator itself, and unless the device used seems obstructive to an exceptional degree, it will be safe to assume the same factor for radiator and valve combined, as given on the charts for ordinary connections.

The effect of reduced working pressure and of throttling on the delivery is naturally greater with an open system than with other methods of piping, since the discharge is controlled entirely by the balance of pressure available at the valve, not by the condensing capacity, which regulates a closed system. An open apparatus, therefore, will have a smaller range of central regulation and the most accurate graduation under normal conditions is easily disturbed by throttling and variation of boiler pressure. For these reasons the claims for

[1] "Gesundheits Ingenieur," Vol. xxix, No. 21.

graduated control are often disputed. Since a reasonably certain reduction of the discharge to a given fraction representing say 3/4, 1/2 and 1/4 of the total heat is the essential feature and main advantage of the open return method, the endeavor should be to design the distributing system with a view to reducing the disturbing effects. This can be done, as previously pointed out, by liberal sized trunk lines, by eliminating as much as possible the uncertain factors that call for throttling, and by lessening the condensation in transit which is liable to vary, and to bear on the volume and resistances to an appreciable extent.

The general procedure in calculating and equalizing an open return system is the same as outlined for the other modes of piping. With the quantities of heat scheduled closely, it will differ only in the degree to which equalization should be carried out. The total pressure loss permissible will be approximately equal to that for a closed system for the same structural dimensions. The total will define the working pressure above atmosphere. In any case, the mean pressure to be taken as a basis for calculation is not likely to exceed 1 lb. and the diagram for that pressure will be found accurate enough.

Example of an Open Return System.—The apparatus represented by the schedule (Fig. 23) will permit a water column of 4 ft. in the return at the boiler up to the level of the open overhead main, corresponding to a total pressure drop permissible of $\frac{48}{.192 \times 144} = 1.73$ lb. per square inch. Allowing for increased heat and friction losses in transit when heating up, the boiler pressure might safely be run up to about 1 lb. per square inch. The system is calculated with the view to obtaining the required discharge at each appliance by a close equalization of the pressure losses from the boiler to the radiators with the least amount of throttling. The initial boiler pressure therefore exceeds the atmospheric only by the margin necessary to overcome the friction and local resistances, which in this case is made to approximate 5 lb. per square foot to all points of delivery, and is a small fraction of that which the water column in the return will permit. The valves are of a type having nearly the full area of the pipe connection, so that the discharge velocity is moderate and not liable to cause noise. They contain the means for adjustment of the full opening to effect still closer equalization. The throt-

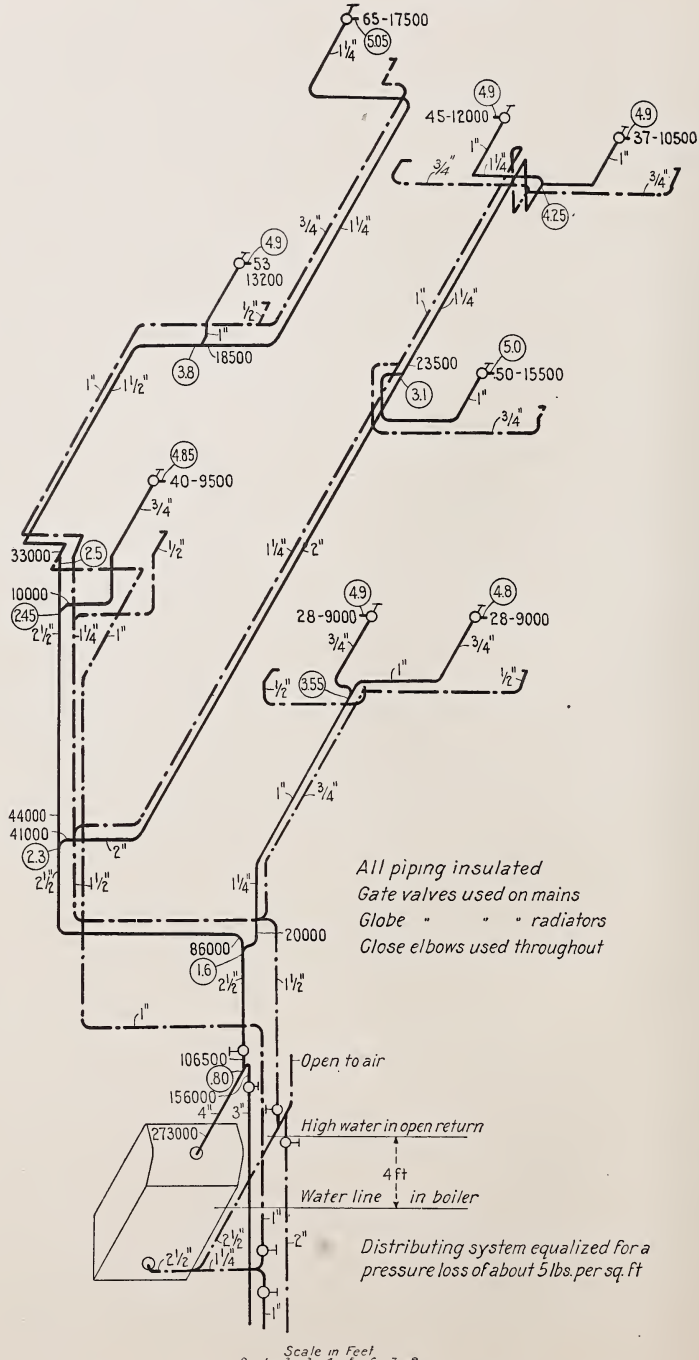


FIG. 23.—Examples of low-pressure steam heating at 1 lb. pressure. Open return system—with adjustable steam valves.

tling incidental to this process will naturally raise the working pressure to an extent depending on the skill of the mechanic, but the plant can be made to circulate perfectly at very low pressure and might properly be termed a vapor heating apparatus. Close regulation of the steam pressure is essential for the successful operation of such a system, since a slight increase will cause an escape of steam through the returns. If this be prevented by automatic trapping, an undue rise of pressure will affect the graduation to some extent, but will not cause any other disturbance, until the water column in the return will seal the system and turn it into a wet return apparatus, poorly designed as such and liable to be noisy. A secondary condition for the successful operation of an open system is therefore a close and positive control over the initial or boiler pressure. Special devices are necessary to secure the proper results.

THE FLOW OF AIR

CHAPTER VIII

THEORY OF THE FLOW

Properties of Air.—The diagram of properties is presented for occasional reference to the weight of the air as the absolute quantity, the charts being based only on volume, which is the convenient measure, and sufficiently accurate in general practice, with the corrections noted.

The diagram also shows the influence of moisture on the volume, which increases rapidly at higher temperatures, and may become of importance in connection with drying problems. The weight of water that can be absorbed by air at different temperatures is given by a separate scale. This weight corresponds by no means to the difference in weight between dry and saturated air, because the volume increases with the percentage of moisture.

The temperature variations enter frequently into the mechanical problems and are accordingly stated and incorporated in the charts used for calculating.

The bearing of barometric pressure and altitude on pressure and power is appreciable and should really be taken into account under extreme conditions. Inasmuch as it is usually negligible and nearly always within the limits of error from other sources, the data has been omitted for the sake of simplicity.

Friction in Conduits.—The general expression for the loss of head by the friction of fluids in conduits $H_f = f\frac{v^2}{2g}\frac{l}{d}$ is applicable to air. The coefficient f naturally takes different values than those for water and steam owing to viscosity and nature of conduits. It varies also with the relation of internal and external friction, that is, upon the ratio of area to contact surface and its roughness.

Weisbach gives the value $f = \frac{.217}{\sqrt{v}}$ for air, derived from experiments. These seem to have been limited to very small tubes with smooth surfaces, and high velocities, entirely outside the

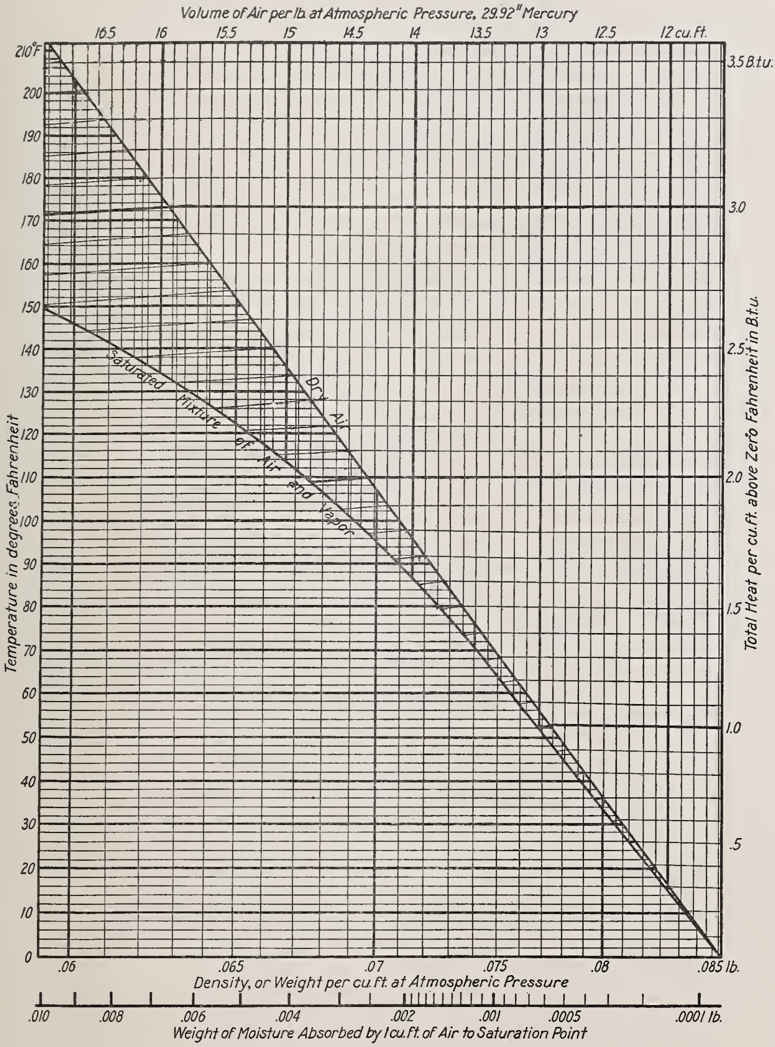


DIAGRAM D.—Properties of air.

range occurring in heating and ventilating practice. According to this formula the coefficient changes only with the velocity.

Grashof finds $f = .0135 + \frac{.001235 + .01d}{d\sqrt{v}}$ thus showing the coefficient to vary both with diameter and velocity.

Pelzer also recognizes these variations including them in his formula for the loss of pressure $P_f = \frac{748}{10^6} \frac{1}{d^{1.373}} w^{2/3} v^2$

Stockalper, as the results of his tests on tunnel ventilating work, gives for cast iron pipe 6 in. to 8 in. diameter

$$P_f = \frac{785}{10^{10}} \frac{1}{d} w\left(5 + \frac{1}{d}\right) v^2$$

being expressed in m/m water column. According to this formula the coefficient increases as the diameter decreases, but is not affected by the velocity of flow.

Lorenz, as quoted by "Die Hütte," stated the friction loss in air pipes to be $P_f = P.f \frac{lv^2}{dT}$, wherein T and P are the mean absolute temperature and pressure in the conduit. The coefficient of friction f is given for diameters ranging from 2 in. to 14 in. and is taken to apply at any velocity.

Brabbée, operating on round sheet metal piping from 12 in. to 32 in. diameter, also used for tunnel ventilation, finds f practically constant for these sizes, at velocities ranging from 10 ft. to 50 ft. per second. His value for f is obtained by a series of very thorough experiments and checks closely with the average of the values by twenty other authorities quoted by him.[1]

Taylor, as the outcome of his tests for the U. S. N.,[2] agrees that f is practically constant for diameters from 12 in. to 27 in. and at velocities up to 100 ft. per second.

Rietschel, experimenting for the Vulcan Iron Works,[3] has compiled a table of values which shows f to vary both with circumference and velocity. Intended to apply to ductwork for ship ventilation, they cover at least partially the conditions maintaining in building practice.

Other formulæ on friction head given in trade publications

[1] "Gesundheits Ing.," Oct. 10 and 20, 1905.
[2] "Transactions N. A. and M. E.," 1905.
[3] "Gesundheits Ing.," July 1, 1905.

such as $p = \frac{lv^2}{25000\,d}$ (d in inches), are mostly simplified variations of those quoted. They include f in a constant which is virtually correct within certain limits.

The results of the various formulæ, when plotted on logarithmic paper, show a fairly close agreement of the lines of diameter and velocity, if charted only within the range from which they are derived and known to apply. The slopes of the diagram presented strike about the average and will be found to give safe values for the friction in fairly well made sheet metal ducts of all sizes and velocities that may occur in ventilating work.

The slope of the lines on the logarithmic chart when expressed in figures, as was done for water and for steam will make the, pressure loss in pounds per square foot.

$$p_f = w.\ 5.13f\,\frac{v^{1.9}}{2g}\cdot\frac{l}{d^{1.18}} \text{ for round pipes}$$

$$\text{and } p_f = w.\ f\,\frac{v^{1.9}}{2g}\left(\frac{c}{a}\right)^{1.18} l \text{ for ducts of any shape.}$$

In this formula w is the density, v, l and d are again the velocity, length and diameter of conduit in feet. The value f is a constant expressing the coefficient of friction in the equation $h_f = f\frac{v^2}{2g}\cdot\frac{l}{d}$ when v = 1 ft. and d = 1 ft., or when $\frac{c}{a} = 1$ for square ducts. The charted average makes it .032 for round pipes, and $\frac{.032}{4^{1.18}} = .00624$ for ducts of any shape. This figure is slightly in excess of most of the experimental data on f available and apply ing to these conditions, also of the results of Weisbach's and Grashof's formulæ for f. It is therefore shown to be conservative and applicable to reasonably well made conduits at any range of diameter and speed.

Factors of Local Resistance.—While the coefficient of friction has been investigated liberally and is known for all sorts of conditions, so as to bring the loss of head from that source well within the limit of error, the factors of local obstruction for changes in shape or direction of conduits, which usually present the greater resistance in ventilating systems, have received comparatively little attention. For many features, even the most frequently occurring, the data published is still incomplete or indefinite.

These various forms of local resistance are not comparable to the standard types or patterns of pipe fittings. They should be designed to suit the requirements of the individual case, but perhaps more often they are shaped to suit the taste and habit of the individual designer. For junctions or breechings, where the relations of static and dynamic head will enter, the factors must necessarily be variable, and cannot be charted, but for other items the values of r can be approximated sufficiently to cover average conditions.

As explained in the general chapter, the loss of head by resistance, which is generally taken to be a function of the velocity head $\frac{v^2}{2g}$, is assumed to vary as the friction head. Accordingly, $p_r = w.r \frac{v^{1.9}}{2g}$, making the lines of resistance parallel to the lines for area, and giving the pressure loss in pounds per square foot for any velocity within the range of the charts.

In order to give the approximate relation of r to $\frac{v^2}{2g}$ which would be correct in the above formula only for $v=1$, when $v^2 = v^{1.9}$, the factor has been taken as a function of the velocity head at 9.25 ft. per second, for which $wr \frac{9.25^2}{2g} = w\ 1.25r \frac{9.25^{1.9}}{2g}$ the constant 1.25 being equal to $\frac{v^2}{v^{1.9}}$. For that velocity, we can write therefore $p_r = w\ 1.25r \frac{v^{1.9}}{2g}$. For other velocities this will not change the results of the formula as given first, but changes the value of r to allow a fair comparison with the coefficients of resistance measured by the velocity head, or quoted elsewhere in standard publications. For higher speeds than 9.25 ft., p_r figures less, for lower speeds more than the factor r indicates.

Simple Forms of Resistance.—The various shapes selected are those most frequently occurring in practice. The factors for bends of different radius represent the average of the coefficients as given by Weisbach, Rietschel, Recknagel and others with an allowance for imperfect construction. They may be applied also to rectangular cross-section, whether the turn is on edge or sideways, as long as the proportion of radius and width or diameter

in the plane of deflection is as stated, excepting for knees and bends in very flat ducts, for which the coefficient increases decidedly when the turn is on edge. In extreme cases the resistance becomes a very uncertain factor. Turns of more or less than 90 degrees may roughly be assumed proportional to the angle of deflection, although this is not strictly correct. If two bends or other shapes deflecting the current follow in close succession, the factors for both may be materially affected, according to the distance between. An offset or S figures more than the turns composing it. For round pipe, elbows are made up in sections, about as shown. The imperfect curvature of these bends is expressed in the factors. Bends of larger radius, are negligible items. Dampers without projecting frames or edges need not be taken into account.

The loss of head by discharge through an orifice in a thin plate, illustrated by No. 1 on Fig. 24 is given for a coefficient of efflux $=.6$. It is assumed that the pressure is lost partly as the dynamic head of discharge, accelerated by the contraction and partly as static head expended in overcoming the resistance of the orifice, the total being expressed again as a function of $1.25 \frac{V^{1.9}}{2g}$, based on the velocity corresponding to the outlet area. The contraction in such an orifice is known to vary with diameter and velocity. The above formula follows these variations at

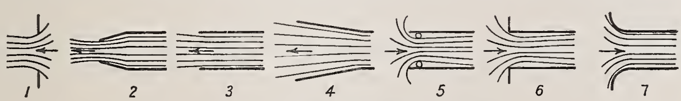


FIG. 24.—Types of inlets and outlets.

least partially. With the stated coefficient of efflux it should give safe values for average cases. This factor does not apply to the frictionless orifice with rounded throat, but only to sharp edged openings in a thin plate, involving considerable contraction of the stream beyond the area of the opening.

Discharging air through a pipe involves a pressure corresponding practically to the velocity head at the points of exit, no matter if the outlet is straight, converging or enlarging, as long as no contraction or eddies occur. The actual variations of this factor for the three pipe ends, Nos. 2, 3 and 4 shown on Fig. 24, do not appear to have been established, but they would probably differ but little from the dynamic head of the discharge based on

the exit area, which is well known to affect delivery to an appreciable extent according to the mouth-piece selected.

Of the three styles of inlets No. 5 and No. 6 present a decided resistance owing to contraction, which are given approximately on the chart for air blast. The factor is based again on the velocity corresponding to the actual area, not of the contracted stream. In No. 7, the ideal contractionless orifice, the two velocities are identical. For an inlet into a pipe, the resistance of this ideal throat is negligible. When used for an outlet from a plenum space, the loss of head is practically that due to the full theoretical velocity, or the entire pressure head, the orifice being the equivalent or blast area.

Equivalent Area.—As stated in the general introduction, the equivalent area is that corresponding to the discharge velocity resulting from the total pressure available at any point of a conduit system. It is the area to which a duct may be contracted without affecting delivery, provided, the contraction and expansion of the stream is gradual and does not involve losses of motion. Any sudden impact and eddies causes a loss of pressure which should be taken into account and subtracted from the total available. The charts will give the theoretical velocity at the intersection of the dotted line with that of the pressure available, and the equivalent area is found along the same velocity line, where it intersects with the ordinate for the volume to be carried.

If this analysis for blast area is made, it is often possible, quite contrary to popular notion, to contract the area of a duct by a considerable portion, without curtailing delivery. Throttling takes place only through a reduction beyond the blast area, or through forms of contraction that will create a loss of motion with consequent decrease of the pressure available and corresponding increase of that area.

The ideal form of contraction with least resistance is simply formed on the lines of natural flow through a sharp edged insertion as shown on Fig. 25, or the lines, on which the "Venturi" meter is shaped, which is known to present but a very small resistance to the flow. The area at the waist in this case might be reduced to the theoretical equivalent, for which $h_{v_1} = H$ and $H - h_{v_1} = 0$, the entire pressure available being converted in passing from a to b into velocity head, and reconverted from b to c into the original relation of static and dynamic head.

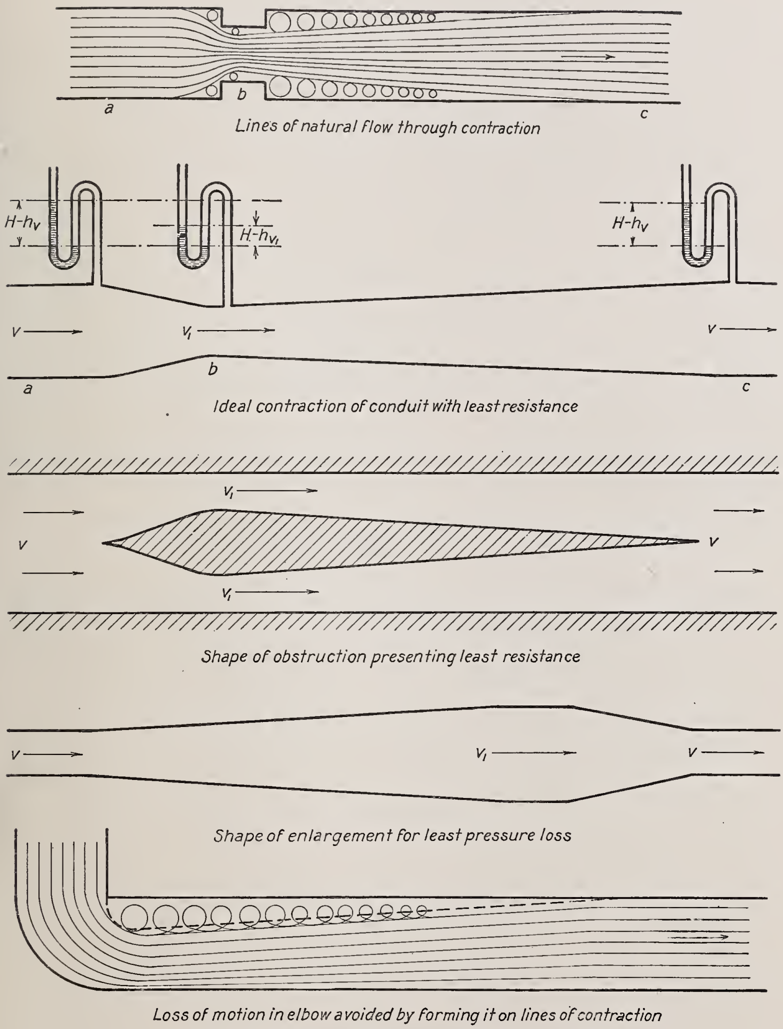


FIG. 25.—Shapes of contraction and enlargement.

Obstructions in conduits that would otherwise curtail delivery can be made to avoid pressure losses and assure a full volume when shaped on these lines.

Gradual enlargements, when built as shown, likewise will obviate losses of motion through impact and counter currents.

It is also possible to reduce the resistance of a sharp elbow in the manner indicated on Fig. 25 by forming it on the lines of the natural flow, involving a contracted area. Loss of motion can thus be avoided through prevention of eddies, and resistance eased thereby, even if no space is available for an open bend. It may not always pay to design ductwork on aero-dynamic lines, but in cases where power must be saved, or dead corners avoided for other reasons, the contraction and enlargement of conduits on this principle is indicated.

As previously pointed out, the reduction should in no case go beyond the equivalent or blast area. This latter is easily determined when a duct system is calculated for pressure loss, the total available being known for any point along the line. It is well to bear in mind when reducing a conduit, that the theoretical area varies between the blower and the suction and discharge ends of the system, being smallest at the former and nearing the duct size toward the latter. A conduit may therefore be contracted to a smaller area at the fan than near the discharge end, according to the actual pressure maintaining at that point and available for conversion into motion.

Composite Forms of Resistance.—All values of r, as given, are necessarily approximate, since they vary in some cases with the direction of the flow as indicated on the charts, and are influenced more or less through other circumstances. No comprehensive rules exist to define these variations. They are appreciable, but generally within the limit of error from other sources and can therefore be neglected in ordinary practice. This applies principally to the composite factors for portions of apparatus, typical and frequently used, which are included for convenience.

For registers, the factor is made up by the sum of losses incident to the changes in direction and velocity in passing from the flue into the room. It is based on the speed in the flue, and is practically the same in either direction. The factor is given for square flue ends used in forced ventilation to secure a straight outflow, and for ends rounded to ease the flow for gravity work. The resistance is not materially affected by shutters or dampers

when these are turned parallel with the direction of the flow, and it is safe to assume the same factor for registers with and without valves. This applies also to registers on the end of a straight run of duct. Registers on the side of a duct usually require some means for throttling off an excess of pressure, in the shape of a narrow throat, or a perforated diaphragm. The resistance of such devices must be made to suit the pressure available at that point of the main. They should be considered

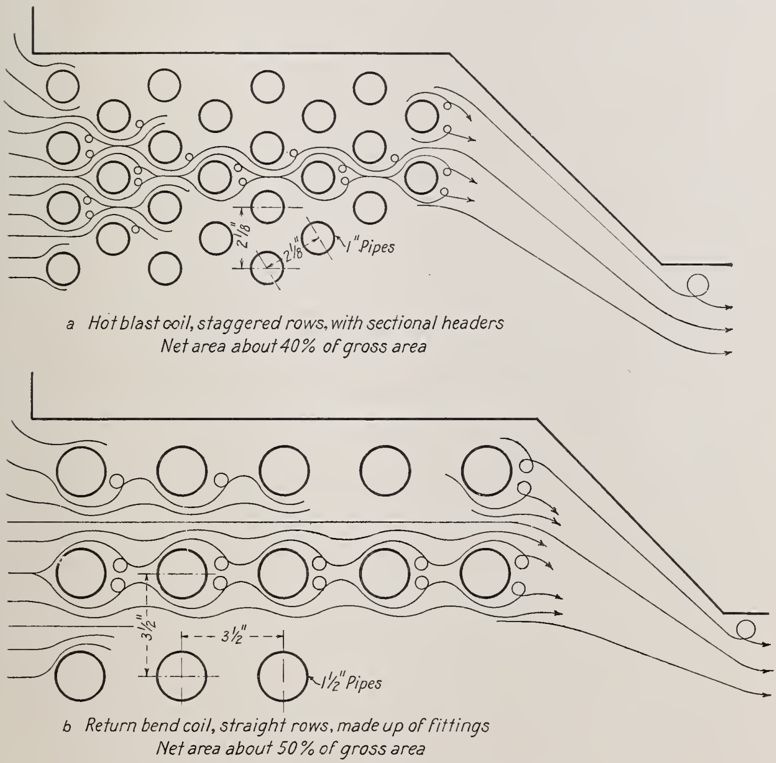


FIG. 26.—Resistance of hot blast coils.

as branch pieces or orifices and figured on the principles developed for them in a separate paragraph.

Heating or tempering coils and indirect stacks offer very considerable obstruction. They usually cause the greater part of the entire pressure loss in forced systems, especially where the velocity between pipes is taken high in order to secure a great heating effect within a small space. It is primarily the relation

of the net area between pipes to the gross area or cross-section of the casing which determines the factor of resistance. The arrangement of pipe heating surface commonly used in connection with ventilating and hot blast work are illustrated by Fig. 26.

Arrangement "a" represents a typical hot blast coil with the pipe rows staggered to increase their heating efficiency. The free area for this style of coil is generally about 40 per cent. of the gross area. If the changes in velocity were not eased by the rounding of the pipes, and the speed between the rows brought down to that for the gross area, the loss of head for each pipe traversed could be expressed by the formula $r = 1 - \left(\frac{a}{a_g}\right)^2$ in this case $1 - \left(\frac{1}{2.5}\right)^2 = .84$. Owing to the close spacing of the pipes the momentum of the flow is maintained from row to row, to an extent which can only be estimated roughly. Moreover, the losses of head by contraction are avoided and eddies are materially lessened by the rounding of the pipes. It is safe to assume that the factor is reduced thereby to at least one-half of the above value, $r = .42$, to which is to be added an allowance for friction, figuring about .03 and making $r = .45$ for each row of pipes. In estimating the factor for the coil as a whole we are to add the extra loss of head after passing the last row and allow for obstruction due to headers and support. For a typical hot blast coil of n rows we may therefore compute $r = .45\,n + .6$. The lines of resistance on the chart are based on this estimate and check as closely as can be expected with experimental data on hand. The recent tests at the experimental station of the Royal Technical Institute,[1] Berlin, for instance, show the pressure losses to be about 20 per cent. smaller, on the average, but for a different spacing of pipes and without fittings, braces and other incidental obstructions.

Type "b" is a straight row coil generally made up with return bends, for indirect heating. The spacing with close headers and open bends will give a ratio for $\frac{a}{a_g}$ of about one-half, making the theoretical factor per row $1 - \left(\frac{1}{2}\right)^2 = .75$. In the

[1] "Mittheilungen der Prüfungsanstalt für Heizungs und Lüftungseinrichtungen," Heft 3.

straight passages the speed of the air is likely to be maintained somewhat more in this coil than in a staggered one, but the loss of motion by eddies is probably greater. Considering these points we are safe in assuming again $r = .5 \times .75 + .025 = .4$ for each row, and $r = .4n + .5$ for the coil as a whole.

It would be impracticable to calculate and give the resistances of the various types and makes of cast iron indirect heating surfaces. Some of the best are no longer in the market and the ideal design, combining the least resistance with the greatest heating effect still remains to be developed. Each manufacturer should test not only the emission, but also the friction loss for his particular designs, under stated conditions, and should present this information in the catalogue or on application. Engineers may then themselves insert this data in the charts for the styles which they are in the habit of using. In the absence of such figures, it is best to estimate the factor of resistance by comparison with similar forms and items of obstruction.

The resistance presented by the casing of a coil or stack bears little relation to that of the pipes or sections and should be considered separately, according to area and shape of inlet and outlet. The inlet, when equal to the gross area as would occur in connection with a chamber, presents slight resistance owing to the low velocity. Connecting with a duct, the velocity head maintaining in the same at the entrance is nearly all lost, unless the enlargement to the gross area of the coil is very gradual. It is safe to assume $r = .8$ to 1. An outlet converging under an angle of about 45° will offer the same obstruction as a cone-shaped mouthpiece for which $r = .6$. When built at right angles, with a round opening opposite the fan inlet r is to be taken $= 1$. The factors for stack casings can be estimated in a similar way on hand of the various forms given on the charts. If one or two sides of the duct are flush with the casing, the factors for entrance and exit may be correspondingly reduced.

Air filters of cheese cloth naturally check the flow of air according to the free area of passage, which depends not only upon the actual surface and the quality of the cloth, but also upon the amount of deposits gathered on the same. The factor is therefore necessarily an uncertain quantity, but it is none the less important to make allowance for this item of resistance. The velocity of passage is generally based upon the filter area or cloth surface. The actual speed between the meshes, by which

the factor might be measured, is much greater. Assuming it to be 10v, and basing the factor on that speed, a fair estimate would be $r = 1$ for loss of speed plus .5 for deflection through oblique passage. For clogging to one-half clear area we may put $r = (1 + .5) \times 2^2 = 6$. This factor checks with figures obtained for filters in fair condition, and approximates the results from the formula given by Rietschel for cheese cloth. For cotton flannel the resistance is from 10 to 20 times greater at the same velocity.

The losses of head at entrance and exit to the filter chamber should be calculated separately according to the respective areas and shapes.

Air washers with baffle plates, also other forms of filters, may be figured approximately as a combination of given factors, or estimated roughly by the losses of headway that may be involved.

Velocity Head.—The third item in the sum of losses making up the total pressure is the velocity head $P_v = w \frac{v^2}{2g}$. It is understood to represent the dynamic head maintained at any point along the conduit which must be the difference between the total P and the static head P_s, the unused part of the resistance head. The charts give its value in pounds per square foot by the special dotted line at the intersection of those of velocity. As an item in the calculation of the total pressure, P_v is to be added only when sudden reductions of speed occur through features outside of those for which the factors are charted. The head-way lost in the items given, for instance that for registers, is always included.

Variations of velocity through changes of area, such as occur at junctions, generally involve a partial conversion of the dynamic into static head, or *vice versa*. Only the losses of motion caused by these reducing or enlarging branch pieces are to be added to the total. Under gradual changes in area, when conversion takes place the losses incidental to it are negligible, but for pieces built indifferently on straight lines with decided angles of deflection, the drop in dynamic head practically expresses the head to be added as a loss.

According to Weisbach and Carnot-Borda, the resistance head involved by sudden enlargements will figure $h_r = \frac{w}{2g}(v - v_1)^2$.

Blaess[1] calls attention to the fact that this formula does not apply strictly to elastic bodies. His tests would indicate that for decided differences of speed, the loss is nearer to $h_r = \frac{w}{2g}(v^2 - v_1^2)$. In such cases it would seem to be safer to use the latter formula, giving the drop in dynamic head, except where this drop is a small fraction of the total, in which case only a part of the stream is dissolved into eddies and the former expression is more nearly correct. For $v_1 = 0$, or $v_1 = v$, the result would be the same from either formula. The values for both can be read from the chart by subtraction of the values for P_v and P_{v_1}, or subtraction v and v_1, which gives the head $\frac{w}{2g}(v - v_1)^2$.

In ventilating practice it is good policy to design conduits to reduce these losses through reductions in speed so as to render them negligible. This can nearly always be done by a taper piece on easy angles. Increases of speed also involve losses of head, but only in so far as taper pieces may cause contraction and eddies. Losses of that character are also easily reduced or practically avoided by intelligent design, bearing in mind that the angles for contraction and enlargement of area should differ, same as the converging and diverging lines of the nozzle with least resistance.

Generally speaking, it will pay to eliminate as much as possible the uncertain factors when laying out a conduit system. This may not be obvious at first, but the fact will force itself upon the engineer whose designs grow out of the calculation.

Junctions.—Unless air is to be delivered against pressure, the head of the flow is usually a large part of the total at any point of a conduit. It is necessary to consider it in problems of distribution.

Branch pieces, breechings, or any other modes of joining two conduits into one, do not present any resistance outside of the portions of bends or the tapers usually connected to them, if designed to conform to the natural direction of the flow as it would result from the static and dynamic pressures at that point. In practice, the theoretical shape to fit this flow can only be approached, and some obstruction is unavoidable, but it should be kept down by appropriate shaping of junctions. This is

[1] Victor Blaess, "Die Strömung in Röhren."

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desirable in order to reduce the total resistance head, as well as to eliminate the uncertain effect on distribution by improper forms of breechings. A fair approximation can always be assured by taking account of the pressure conditions in main and branch which give the throat velocity obtainable, and define at the same time the resultant angle of flow.

If P is the total pressure in the main at the junction, P_v the velocity head, or pressure in axial direction, $P-P_v$, or P_s is the static head or radial pressure in excess of the atmospheric. Calling P_b the back pressure caused by resistances in the branch, the radial outflow would be governed by P_s-P_b or $P-P_v-P_b=P_{v1}$. The diagonal of the parallelogram resulting from the velocities due to P_v and P_{v1} gives the theoretical discharge velocity and its direction. Fig. 27 illustrates the aerodynamic conditions involved.

When the two forces act at right angles, as is the case on a straight run of main, the sum of squares of the two velocities equals the square of the resultant or actual speed of discharge. Since the pressures vary also as the square of velocities, we have, accordingly, if v_2 designates the actual speed in the branch,

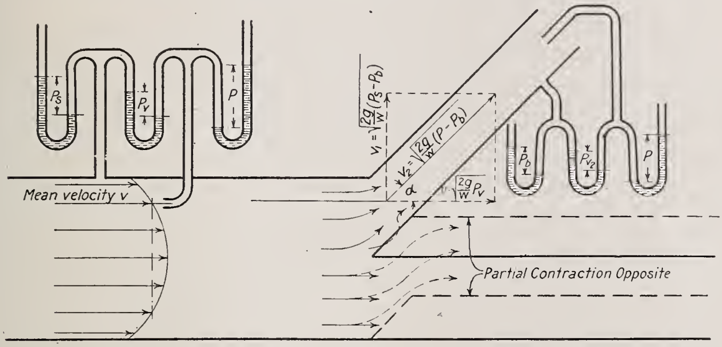
$$P_v+P_{v1}=P_{v2}, \text{ and } v_2=\sqrt{\frac{2g}{w}(P_v+P_{v1})}=\sqrt{\frac{2g}{w}(P-P_b)}$$

It follows that the ratio $\frac{v_1}{v_2}=\sin.\alpha$, $\frac{v}{v_2}=\cos.\alpha$ and $\frac{v_1}{v}=\text{tang.}\,\alpha$, α being the angle under which the discharge takes place. Hence we have $v_2=\frac{v_1}{\sin.\alpha}=\frac{v}{\cos.\alpha}$. Thus the natural direction of discharge is given by the relation of velocity heads for main and branch. Either the angle α should be suited to this ratio, or the ratio may be suited to the angle by an increase or reduction of back pressure on the branch which controls the velocity in the same.

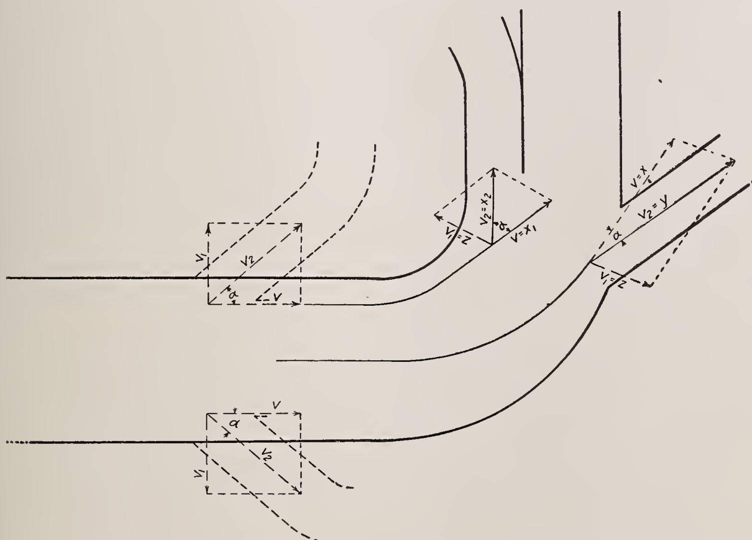
If the branch, measured at right angles to the line of v_2 gives a lower velocity than the theoretical, there will be a sudden reduction of speed, or loss of motion, which means loss of head. If the throat is smaller, the volume is necessarily reduced, but only about as the square root of the area, since the back pressure is relieved as the flow is lessened. When the angle of the junction is chosen too large or too small there is a loss of head due to eddies and impact which is difficult to estimate.

In extreme cases it may use up all the head available, and even cause reverse action. Hence the importance of considering the form of breechings.

Strictly speaking, the back-pressure P_b is purely that by fric-



Velocity in Branch taken from a Straight Main



Velocity in Branch taken from a Curved Main

FIG. 27.

tion and obstruction. Whatever dynamic head is represented by the discharge itself, and by sudden reductions of speed are to be considered as making up the velocity head P_{v_2} resulting at the throat, which is being spent along the branch. In the

case of gradual enlargement of area with reconversion of motion into pressure, the gain of static pressure is to be subtracted from from P_b and added to it, when pressure is released into motion through a gradual contraction, or when extra head is needed to create higher speed of discharge than that maintaining at the branch piece.

When the conditions defined above are met by design, the branch piece reaches its greatest capacity, or full efficiency, each shape and angle having its characteristic curve of discharge with the high point at a certain ratio of P_v and P_{v_2}. This has been neatly demonstrated by Taylor's experiments for the U. S. N. previously referred to. At full capacity, for the ideal breeching, P_v^b represents purely the back pressure beyond the junction itself, P_v and P_{v_2} the velocity heads at main and branch. Since there is no loss by eddies or otherwise, $P = P_s + P_b + P_{v_2}$, or $P_v + P_{v_1} = P_{v_2}$.

In order to get the full effect of v as a component of v_2, a branch piece should be built to face the current. In other words, the connection should form the offset or reduction on the main, as indicated in Fig. 27. When the outlet is flush with the side of a conduit, even though the angle be taken correctly, there will be an increase of speed on the contracted side of the main opposite, with a deflection toward the side of the branch, but we cannot figure on the full dynamic head of the main. Only a portion of it, depending on the shape of the breeching as a whole, is utilized. It may be estimated roughly from the area facing the current. Losses of motion caused by incorrect branch pieces also affect the forward flow through the main. Wherever possible they should be made a negligible item by appropriate design of ducts.

Abnormal conditions will prevail when the flow within the main is deflected by curves, as illustrated also by Fig. 27. The outflow is evidently governed by several factors. Branches on the outer side are favored by the direction as well as the strength of flow, the increase of actual velocity depending on the radius of curvature and the exact location of the junction. On the "lee" side, the reverse takes place. When the static head is low at that point, the eddies in the main may create a suction on the branch. It is difficult to compute the discharge under such conditions. Branches on curves should be avoided. It is best to join ahead of a turn in the main, not after a turn, since the flow is always disturbed beyond for a length of several diameters.

The ratio of static and dynamic head may vary to a considerable extent, between the junctions of the same conduit. It differs also according to the method of distribution, that is, between plenum and high speed. Since the shape of the branch pieces will not only affect the resistance, but also the effective area of the connection it is essential to select a form of junction that will at least be approximately correct.

It will be clear from the foregoing argument, that form "a" in Fig. 28 is proper for an outlet from a plenum chamber under sufficient pressure to overcome the extra resistance due to the contracted current at the orifice. Outlet "b" avoids this loss by contraction and should be applied when highest velocity is to be obtained from a given pressure. For plenum chambers $v=0$, and $\alpha=90$ degrees. Hence it would be as disadvantageous to discharge under smaller angles as it is to use "a" and "b" on ducts with air in rapid motion. The wrong way is shown by the upper figures.

The type of outlet illustrated by "c" is correctly applied when the pressure head in the main would result in greater velocity than is desirable, the branch being too short to create back-pressure by friction. In this case the extra head is all converted into motion, the excess of which is lost in eddies after passing a perforated diaphragm. This form of throttling may be used also on mains with moderate velocity, but the effective area should then be measured across the natural angle of discharge. Diaphragms should not be used, however, where the static head is very low, or the velocity high. In such cases a short branch leading off at a small angle offers better opportunity of equalization.

Form "d" is for high-pressure and low velocity on the main, or for slight back-pressure on the branch. For $\alpha=60$ degree, $P_{v1}=3P_v, P_{v2}=4P_v$ and $v_2=2v$. This maintains, of course, only for perfectly shaped reducing and branch pieces. The forms "e" and "f" are in order where $P_{v1}=P_v$, and $P_{v1}=.33\ P_v$, hence $P_{v2} = 2\ P_v$ and $P_{v2}=1.33\ P_v$ or $\alpha=45$ degrees and 30 degrees respectively, making $v_2=1.41$ v and 1.16 v. When the velocity in the main is high, or when it is desirable to keep it lower for the branch, still smaller angles are advisable, $\alpha=22\ 1/2$ degrees will make $P_{v1}=.17\ P_v$, $P_{v2}=1.17P_v$ and $v_2=1.08v$.

Near the end of a main, where little pressure head is left, the highest velocity obtainable for the last branches as a rule does not exceed materially that of the main. It follows that the angle of

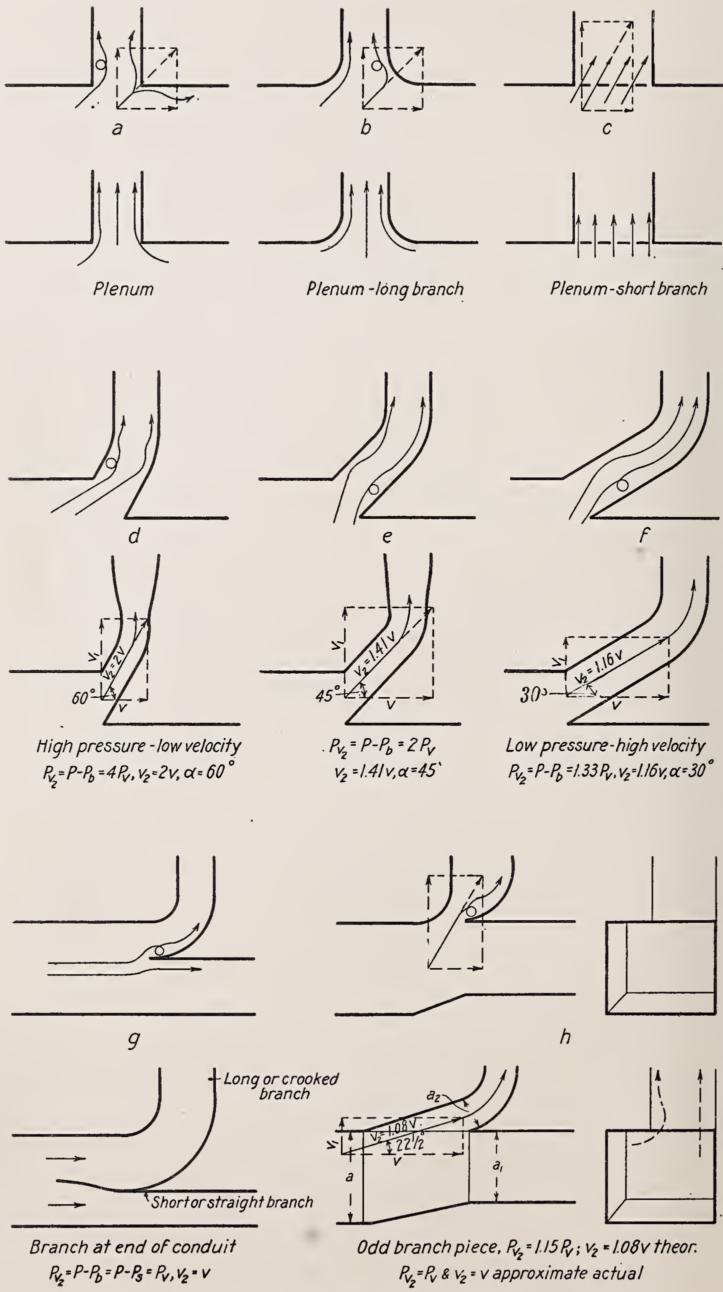


FIG. 28.—Types of branch pieces.

discharge must be as small as possible. The branch pieces take the form of breechings, as shown by "g". The same form is proper when the resistance of a branch calls for lower velocity. In such cases the throat should be sized for the speed in the main. Any enlargements of area that may be necessary to equalize resistance beyond the breeching should be made gradual, so as to convert speed into pressure that may be available for overcoming resistances. In any event, it is important to proportion the throat areas to prevent contraction and eddies at the junction, with consequent uncertainty of distribution.

Very frequently, branches are to be taken off on the straight side of a conduit, the reduction of size being on the top or bottom, or on the opposite side. In order to reduce the pressure loss in the taper piece, it should be built on easy slopes, with the branch starting also at a small angle wherever possible, about as shown by form "h." If the branch creates a back-pressure suited to this angle, the loss by eddies will be slight, and the velocity obtained in the branch is practically that of the main. The combined area of a_1 and a_2 can be made about equal to a for any branches at less than 30 degrees. The area of branch pieces built in this style is clearly defined, while for branches starting with a throat as shown by the alternate to "h" the effective area is difficult to determine, especially when the branch offers little resistance, so that the natural outflow does not follow the lines of the throat.

Type "h" with modifications is applied in a great number of cases. Starting with small angles, say up to 22 1/2 degrees, it presupposes a back pressure on the branch nearly equal to the static head, and a theoretical branch velocity of not over 1.08v. The loss of head caused by the breeching itself reduces P_v and P_{v_1} by a fraction, hence also P_{v_2} and the resultant v_2. For a fairly well designed branch piece P_{v_2} therefore, closely approaches P_v. If we assume $P_v = P_{v_2}$ the loss entailed by the breeching will be $\sin.\alpha^2 P_v$, which figures $.15P_v$ for an angle of 22 1/2 degrees. The calculation of such a branch piece then reduces itself to the simple problem of making its back pressure, that is, its total resistance by friction and obstruction, equal to the static head, less the estimated loss in the breeching. Since the factors for outlets, registers and other items include also the incidental losses in dynamic head, it is best to add also all other losses of velocity head on the branch, thus making its total pres-

sure come up to $P-.15\ P_v$, or whatever fraction of P_v is estimated to be lost in branching off.

If the pressure losses in the branch and junction are thus equalized to make up P, it is safe to assume that v_2 approaches 2v for sharp angles of 60 degrees, 1.41v for 45 degrees, 1.15v for 30 degrees, and will be about equal to v for branches starting in a forward direction. The throat velocity should never be less than the speed in the main, except for junctions on the inside of curves, or when poor shape of connections cannot be avoided.

The back pressure, as a rule, can be regulated by the length for which the size of the throat is maintained, and by obstructions. Thus it may be desirable to enlarge directly beyond the junction, or it may be necessary to keep an even size up to the outlet. If that does not offer sufficient resistance, the velocity as well as the angle of outflow from the main are to be increased.

The losses of head in the breeching, for the main itself, are not material when branch pieces are designed to reduce deflection of current, and to avoid sudden changes in speed. In taper pieces built on easy angles they are negligible. Ill-fitting shapes with sharp angles, however, are decided obstructions. Deflectors in the form of "scoops" for certain branches will also seriously impede the flow and reduce the head available for outlets beyond. Such protruding scoops are make-shifts showing the lack of intelligent design. The correct shapes are developed by calculation.

In general, it may be said that a system of air conduits for heating or ventilating purposes should be designed with a view to reducing all sudden changes of speed in order to eliminate the incidental losses of motion, which increase the uncertainties of distribution. The greatest *continuity of motion* with gradual changes in velocity is always more successful and often more economical.

The same general principles as outlined for mains and branches under pressure, will also apply to conduits under suction, but it should always be borne in mind, that angles for enlargement of area, or reduction of speed should be smaller than those for contraction or increasing velocity.

Total Pressure.—The auxiliary chart is to be used for the purpose of ready determination of the theoretical speeds, the blast area and the power needs. It gives a larger range of pres-

sures, representing P, the sum of P_f, P_r and P_v, for a conduit system, in pounds per square foot, which is the natural and most convenient measure for such calculations. For conversion into ounces per square inch and inches of water column, separate scales are given.

For air blast at high velocities and for certain classes of ventilating work the pressure P is always created by mechanical means. The blowing machines generally used for that purpose impart their power to the air in the form of motion and compression or static and dynamic head. Theoretically, propeller (axial) as well as centrifugal (radial) fans can be designed to yield pressure and velocity in almost any ratio and intensity, using the right proportions, curvature of blades, housing, and number of stages. The commercial types of single stage axial and radial fans cover a fairly wide range of conditions and will serve well enough in ventilating practice, when intelligently applied. Whatever economy might be secured by specially built machines would rarely pay for the additional investment.

According to Murgue's theory the ideal centrifugal fan wheel with radial blades would produce a total pressure equal to twice the velocity head due to its peripheral speed. The resistances by friction, the losses of motion by impact in wheel and casing, also the short-circuiting by leakage around the rim bring the net pressure obtained by the ordinary type of blowers down to about one-half of the theoretical, making it simply equal to the head due to the peripheral speed.

The velocity at which the air leaves the wheel is not identical with the tip speed, but is a component of tangential and angular speed. Blades curved backward will give relatively more static head at lower velocity of exit. They are indicated for blowers delivering against plenum, at moderate initial speed. When curved forward, the exit velocity will be greater than the tip speed of the blades, which is more advantageous for a free discharge or moderate back pressure. With the curvature of blades and proportions adapted to create about the right relation of dynamic and static head, the net pressure will be somewhere between the velocity head of the tip speed and the theoretical. The aim in selecting the right type of wheel should be again to secure the greatest practicable *continuity of motion.* To this end the unavoidable drop of velocity from the blade ends to the duct should be kept down by appropriate angle of exit, by

the size of outlet and the shape of housing. The latter should be designed to convert the excess of velocity into pressure without undue losses. It will not pay, however, to resort to extreme curvature of blades which would require stationary guide vanes and keep down the aerodynamic efficiency of the machine by increased friction. Small angles of exit also reduce the effective diameter of the wheel as measured across the outlets formed by the inclined blades. Where the speed of exit cannot be brought down to approach that of the discharge duct, a tapering outlet on the casing will reduce the dynamic head at a moderate loss. For the same reasons, fans without suction ducts should have large inlets, or funnel-shaped throats to secure a more gradual increase of speed. Two inlets will reduce the resistance of entrance into the fan owing to the much smaller velocity, and thereby increase the delivery.

When the commercial fan does not permit close adaptation to the static and dynamic heads desired, or when it is likely to give low efficiency for other reasons, the peripheral velocity should be assumed from 5 to 10 per cent. greater than that due to the total pressure. On the other hand, when all the factors are known, and the blower can be selected intelligently, it is safe to take the tip speed from 5 to 10 per cent. smaller than the theoretical.

The same principles will apply to other designs, such as the drum type and the pressure blowers, but the relations of tip speed and theoretical pressure will differ. As volume blowers the former give more dynamic head, and with blades curved forward the wheel may be smaller. The charted data as to speed and blast area is intended only for the ordinary type of centrifugal wheel.

With propeller fans the static and dynamic pressure created will depend largely upon the slope of the screw, giving a greater or smaller relative motion. The velocity theoretically obtainable would be $v = \text{tang. } \alpha$ wherein α is the angle of inclination. With 45 degrees at the circumference, this velocity, for a perfect screw operating without back-pressure, would be equal to its peripheral speed. In practical designs of such fans, only a series discs, imperfect portions of the ideal helix, are used. Under the best conditions, with free inlet and discharge the theoretical velocity is only approached near the circumference, hence the low efficiency when operated against pressure. To create pressure, the angle of inclination should be reduced, giving less forward motion and

more power to compress. The less effective portions of blades near the center can be cut off by cones at the inlet and outlet, guiding the air along easy lines, and with gradual increase of speed, toward the revolving discs.

Fans of all types should be designed or selected strictly to suit the conditions of load, that is, to produce a certain pressure difference equal to the sum of resistances $P_f + P_r$, while giving approximately a desired velocity head P_v at the outlet and making up together the total pressure P. The fan capacities, under working conditions thus established, should not be estimated from those obtaining under free delivery or through the theoretical orifice as given sometimes in the makers' catalogues. Either mode of rating is liable to be deceptive, since the actual discharge depends both upon the pressures to be overcome outside and the efficiency within, or the closeness with which the requirements are met by the machine. The correct way to select fans would lead through a study of their characteristic curves under varying conditions of load. If such data is not available it is best to call for a fan that will maintain a differential pressure P, while discharging the volume at about the desired velocity, which velocity should approach that corresponding to the equivalent area at the outlet, as figured from the back-pressure on the fan. The manufacturer who knows the characteristics of his machines will then be in the position to select the most advantageous type and size. The manufacturer who does not know these characteristics, will be handicapped, either by competing with a machine that is too large, or by having to replace one that will not come up to a clearly specified performance.

For heating and ventilating by gravity, the total pressure must be produced by differences of temperature or weight, the motion being produced by that portion of the pressure not consumed in friction and local resistances. The total pressure available is usually limited by various conditions. The charts give the ready means to determine pressure and velocities. A guide to their application will be found in a special chapter.

Frequently, the head to be created by mechanical means is reduced or increased by differences of temperature. It is well to consider the effect when that disturbing factor is likely to bear on the delivery of a fan under normal conditions, since the heat may sensibly influence the pressure to be maintained and modify the design, the speed and the power requirements.

The charts for gravity work give the pressure to be added or deducted.

Motive Power.—The theoretical power in foot-pounds to move a given volume of air against a certain resistance can be expressed as the product of the volume in cubic feet per second and the pressure in pounds per square foot. In h. p. it will be $\frac{QP}{550}$.

This is the power that must be expended in overcoming the resistances P_f and P_r outside of the inlet and outlet of the fan and in creating P_v. The energy absorbed by the aerodynamic losses within the fan, also the friction of bearings, and of belts or gears, must be added in order to arrive at the actual power to be supplied by a motor. For a fan efficiency of 50 per cent., the motor should be designed to develop twice the power of blast. When lower efficiency is likely, it should be further increased.

Granted, that the total pressure P has been determined accurately, with due consideration of all resistances, and corrected for lower barometric pressure, higher or lower temperatures, and any buoyancy or depression due to unbalanced air columns, and granted also, that the fan selected is doing its best work at the intended speed and pressure relations, the efficiency of the several commercial types will vary but little. A motor of twice the power should be ample under these conditions. For belts or chain gear, speeds below or beyond the point of highest efficiency, and other adverse factors, extra allowance should be made.

CHART VIII

AIR BLAST AT HIGH VELOCITIES

THROUGH ROUND SHEET METAL CONDUITS

At a Mean Temperature of 70° F.

Pressure in lb. per sq.ft. to create velocity $p_v = .075\frac{v^2}{2g}$. $v = \frac{Q}{a}$.

" " " overcome friction $p_f = .075 \times .032\frac{v^{1.9}}{2g}\frac{l}{d^{1.18}}$.

" " " " resistance of obstruction $p_r = .075 \times 1.25r\frac{v^{1.9}}{2g}$.

Total pressure required $P = p_v + P_f + P_r$ (net between fan inlet and outlet).

Power to move Q against P in h.p. $= \frac{QP}{550} = \frac{1}{2}$ motive power (approximate).

Theoretical velocity to create P in ft. p.s. $= V = \sqrt{2g\frac{P}{.075}}$ (approximate tip speed of ordinary centrifugal wheel).

Blast area of fan or theoretical outlet area in sq.ft. $A = \frac{Q}{V}$.

Diameter of fan wheel in ft. $D = \frac{60V}{n\pi}$ (approximate).

Revolution per minute $n = \frac{60V}{D\pi}$ (approximate).

Corrections

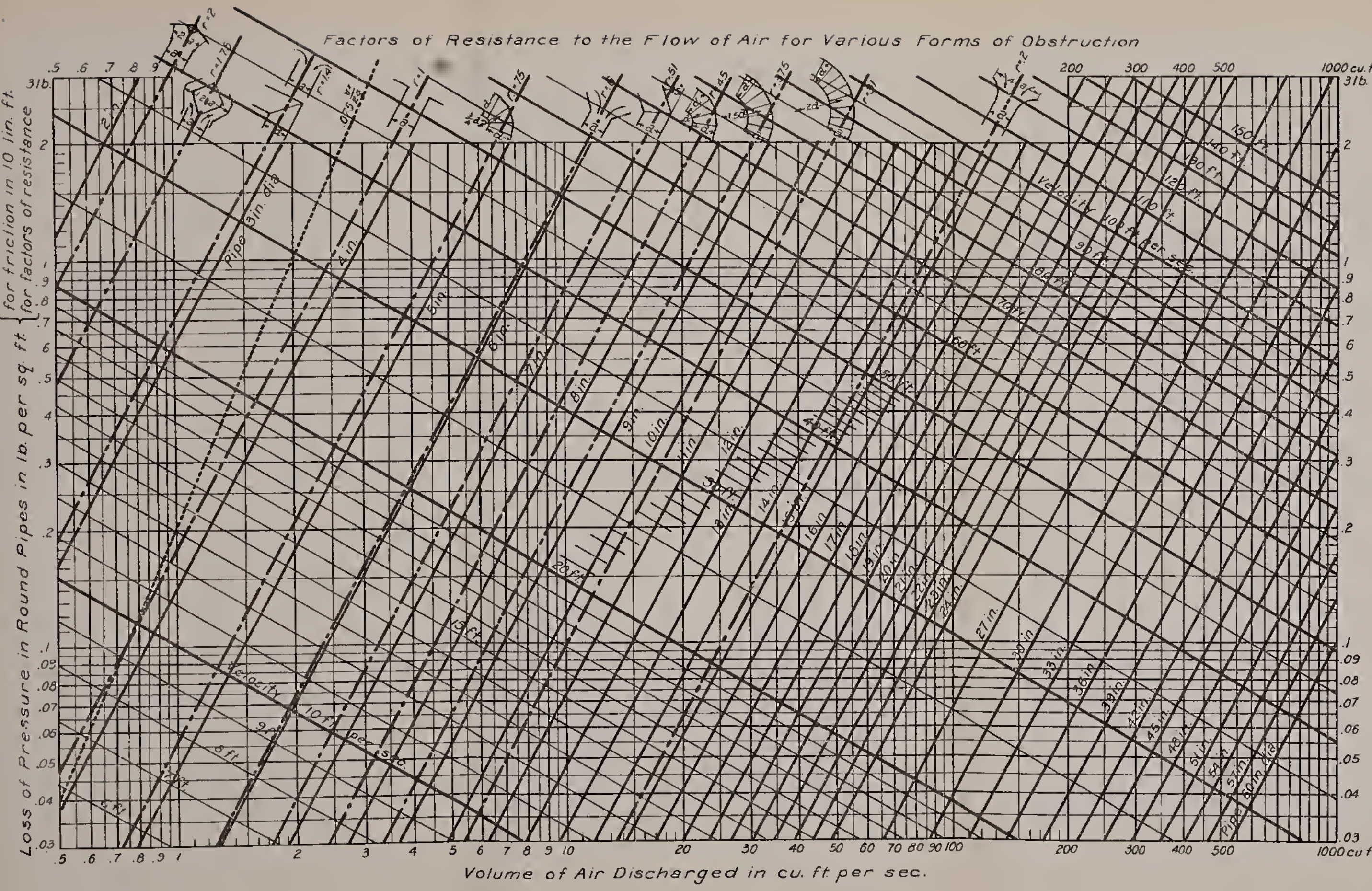
For leakage add to the volume from .5% to 1.5% per 10 lin.ft. according to pressure, character of conduits and ratio of $\frac{c}{a}$.

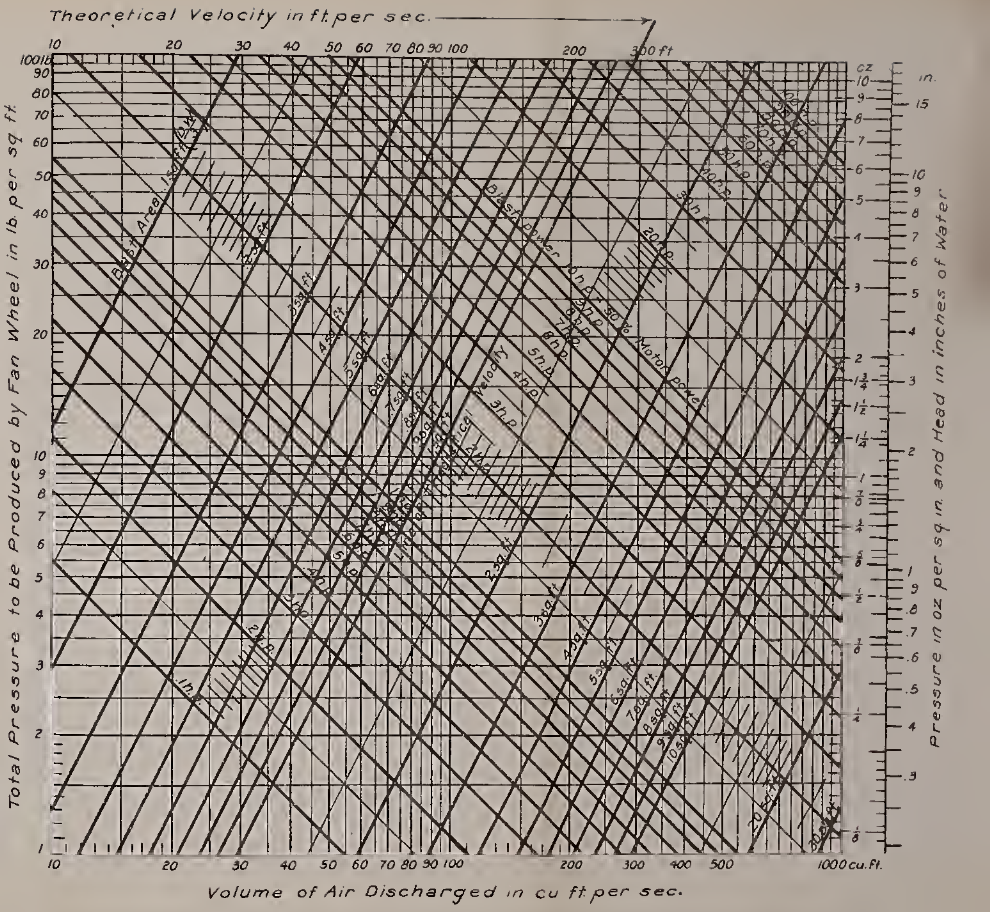
For deformed and poorly built sheet iron piping add from 10% to 30% to friction head.

For air temperatures below or above 70° F., correct pressure and horse power by adding or subtracting 1% for every five degrees of variation.

For square, rectangular, or odd shapes of cross-section with equal area multiply the friction head by $\left(\frac{c}{d\pi}\right)^{1.18}$.

Factors for square and rectangular shapes $\begin{cases} 1\times1 = 1.15 \\ 1\times2 = 1.23 \\ 1\times3 = 1.36 \\ 1\times4 = 1.50 \\ 1\times5 = 1.65 \end{cases}$





CHAPTER IX

AIR BLAST AT HIGH VELOCITIES

Chart VIII.—The chart for air blast covers a range of conditions met in a great variety of apparatus, such as mine, tunnel and workshop ventilation, also blast and suction devices for industrial purposes. It is plotted for round sheet metal conduits usually employed for such work up to any pressure differences handled by the commercial centrifugal blowers. The lines of velocity, diameter and pressure are based on the formula given in the general chapter on the flow of air as applied to round pipes. For square ducts the data will be found on Charts IX. and X. To follow general practice, the diameters are stated in inches, in place of the area. The friction head is charted for a length of 10 ft. The factors for local resistance are those most frequently met in this class of work.

Pressure and h. p. are calculated on the basis of $w = .075$ which is the weight of air per cubic foot at 70° F., at average barometric pressure. For warmer or colder air to be kept in motion, without change in temperature and volume, as measured in transit, the pressure and h. p. increase directly as the density and inversely as the absolute temperatures. The corrections to be made are stated.

The variations in density due to the compression by fans and to changes in humidity, within the range of conditions likely to occur, are very slight and may be neglected in ventilating practice.

The frequent corrections to the friction head, to be made for leakage and imperfect conduits, and the factors for square and rectangular cross-section, are also given. For odd shapes, with equal area, the friction head is to be multiplied by $\left(\frac{c}{d\pi}\right)^{1.18}$, but the increase of area to be made for flatness is only proportionate to the square root of the factor for the friction head.

The auxiliary chart is arranged to determine the power requirement, blast area, and tip speed of fans from the total pressure,

as derived from the calculation. Separate scales are given for ready conversion of the pressure in pounds per square foot into ounces per square inch and inches of water column.

Outline of the Problem.—The simplest proposition in air blast is to convey a stated volume through a conduit of given length. Sometimes the velocities of intake or discharge, or at other points are predetermined, or the power available for creating the flow may be limited.

For a given length and pressure drop the volume varies approximately as the square root of the fifth power of the diameter, and for a given diameter and head, it varies inversely as the square root of the length. To the nominal length should be added the equivalents in resistance of any obstructions on the run. It is most essential, therefore, to consider length and obstructions, and their relation to pressure and density. In other words, we should always take into account $\frac{P}{w.l}$ or $\frac{H}{l}$, the ratio of height to length, or what may be called the slope of a conduit, as well as the hydraulic (in this case aero-dynamic) radius $\frac{a}{c}$, or the diameter for round piping. Calculations based on area and initial pressure alone are liable to be grossly inaccurate.

For a simple conduit the problem then resolves itself into the calculation of the diameter and velocity when pressure and length are given, or into determining the pressure, if diameter and speed are fixed conditions. In either case the unknown factors can be obtained easily by chart. The equation is solved when all the items for P_f, P_r and P_v, are counted up, and the sum P agrees with the pressure figured for the given or desired areas and velocities.

The situation becomes more complex when stated volumes are to be delivered or collected through a system of conduits branching off to several points. The total pressure to be maintained by the fan is usually made up of the suction at the inlet and the excess pressure available at the outlet. If the air is taken from and discharged against the atmosphere, the difference of pressure between the fan inlet and any intake, and between the fan outlet and any point of delivery, must necessarily be the same. In other words, the sum $P_f+P_r+P_v$, when counted up from the fan forward or backward, will always equalize itself for any number of discharges, no matter what may be their size and distance from

the impeller. Hence the branches must be proportioned so as to present an *equal resistance while delivering the desired volume.* In its elements, the problem is the same as for a single conduit, but multiplied by the necessity to figure and equalize resistances to several points for a common pressure head. It is further complicated occasionally by back-pressure or suction, affecting individual points of delivery or differences in buoyancy due to heat, which must be balanced.

It may be questioned, whether it is practicable to equalize a complex system of conduits, if all the calculations are to be made by formula, but the method indicated in the following chapters with the aid of the charts should simplify the process and make it pay, mainly through greater assurance of results.

Application of Chart VIII.—A scale plan in the rough of conduits for air blast or suction may be used as a schedule on which the volume of air in cubic feet per second to be carried in such portions is to be marked. The preliminary sizes of mains are best determined on the basis of an even rate of resistance. Assuming first an initial velocity for the main, as experience may dictate, we can find the friction head per 10 lin. ft. at the intersection of the corresponding velocity line with that of the diameter. If this gives a pressure loss of, say, .4 lb. per square foot, the reduced area of the main after each branch is found directly on the same horizontal line, that is, at the same rate per unit of length, for any number of cubic feet to be carried. In this way an entire system can be proportioned quickly with the velocity decreasing in a gradual and natural way. The sizes so obtained should be marked on the schedule, those for the connections and branches being of course only tentative and subject to correction as equalization will demand. For long duct systems, when the pressure must be kept up, the rate should be small, while in other cases, where space and first cost are limited, it may be necessary to work under more friction.

The pressure losses for the entire system can now be computed from the chart on hand of this schedule. Beginning at the furthest or least favored point of discharge, the various items are added up for each individual run, that is, for the lengths within which the volume is constant. The sum of losses, when marked at each junction, will give the difference of pressure between it and the atmosphere, the total at the fan representing the pressure

against which the air is to be delivered. The intake, similarly figured, will give the amount of suction.

The intermediate branches, nearer to the fan, shorter, or in favoring location, when proportioned by thumb rule, will invite the flow along the path of least resistance, that is, the short, straight branches offering less friction and obstruction will discharge more than intended. The greater tax on the main caused thereby increases the pressure loss toward the far end of the line and involves a reduction of discharge further away. Hence the necessity of equalizing the resistances through the branches. This can be done by modification of the sizes, shapes, length, and features of obstruction. One or two trials generally suffice to bring the resistance of a branch close enough to the pressure available.

In equalizing the various branches it is important to take note of the static and dynamic head prevailing at each junction, in order to select the proper form of breeching or to estimate the resistance for a given type. Although the round piping does not lend itself as readily to the correct shaping of the junctions, in a general way, the branch and reducing pieces can be, and should be, adapted to the static and dynamic heads according to the simpler rules developed in the chapter on junctions.

The figured schedule will finally give the total pressure difference to be maintained by the fan. The size of blower and motor may then be determined, on hand of the auxiliary chart. At the intersection of the lines for the total volume, and the total pressure against which it is to be moved, we find the theoretical blast area and the blast power. The intersection of the line of total pressure with that of theoretical velocity will give the latter, which is approximately the tip speed necessary to create the required pressure with the ordinary type of blower. The size of wheel being defined by the blast area, this tip speed determines the revolutions per minute as per formula given on the chart. If the blast area of a standard size does not fit, the width of the fan should be changed. Large volumes to be moved against low resistancee thus will call for relatively wider wheels, and vice-versa. In cases where the width cannot be modified, the speed may be varied enough to obtained the desired volume, but at lower efficiency.

Examples of Air Blast Arrangement.—Fig. 29, A and B, represent two distinct tendencies in the design of blast piping. One

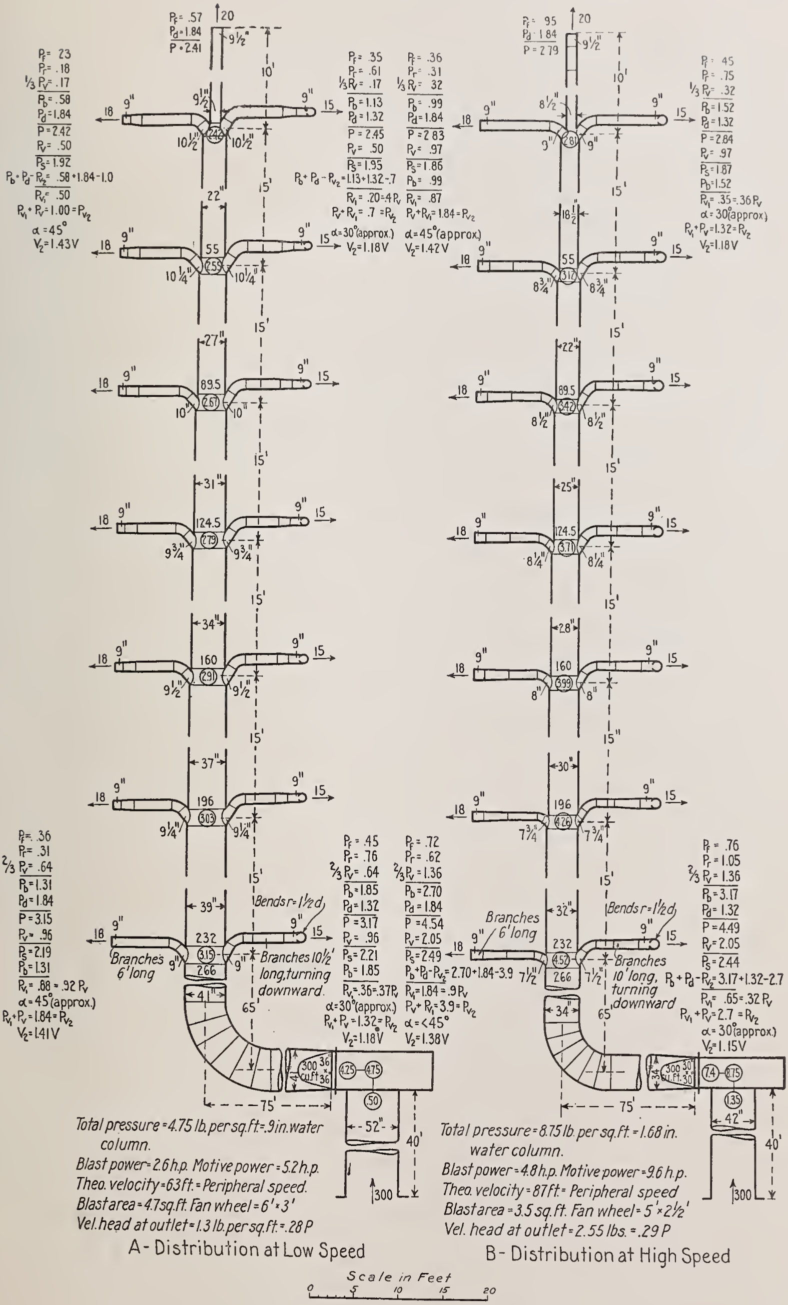


FIG. 29.—Examples of blast arrangement.

Outlets on one side to discharge 18 cu. ft. at 2400 ft. velocity per min.
Outlets on other side to discharge 15 cu. ft. at 2000 ft. velocity per min.
Outlets on end to discharge 20 cu. ft. at 2400 ft. velocity per min.

is proportioned with the idea of saving in first cost through reduction of conduits and size of fan, while the other one is designed with a view to economy in operation. The proper method for any particular case is indicated by circumstances which may point one way or the other, but aside from the power question, the idea illustrated by example "A" will be found preferable on account of the smaller motor, which partially offsets the greater cost for fan and conduits. It will also give greater assurance of good distribution and smaller fluctuations under throttled delivery.

As will be noted, the net performances are equal, the same volumes being delivered with the same velocity at outlets. The lengths of conduits and features of obstruction are identical. With the plan "A" the velocity is kept lower in the main, which approaches the function of a distributing drum or plenum chamber. Plan "B" on the other hand, is sized for higher speed, which involves also a proportionately greater dynamic head.

Since a uniform diameter of main rapidly decreases the ratio of velocity and pressure heads, and a uniform velocity increases the resistance for the smaller sizes, the mains, in either case, are proportioned by the method outlined, for an even rate of pressure loss, which gives a gradual and natural decrease of speed. The ratio of dynamic and static head varies also from junction to junction, and for the same length of branches the angle of the breeching and its throat diameter ought to be graduated to conform. In order to simplify construction, the angle is assumed to be uniform for each set of branches, being made 45 degrees for the short ones with less resistance, and 30 degrees for the opposite row, presenting more back-pressure, and the resistances are varied to conform to the angle, by graduations of diameter. The initial speed v in the main for scheme "A" is figured from the size of the first branch, the diameter of which is made to correspond to the prescribed discharge velocity v_2. For 45 degree angle, $v = \frac{v_2}{\cos 45^\circ} = \frac{v_2}{1.41}$. This initial speed defines also the rate of pressure loss, about .05 lb. per 10 lin. ft. The decreasing sizes for each sectional run are taken from the chart along the same horizontal line. For scheme "B" the last branch has been taken as a basis, the final velocity in the last section of the main figuring again about $\frac{v_2}{\cos \alpha} = \frac{v_2}{1.41}$. The rate of pressure loss in this case is

about 0.15 lb. per 10 lin. ft. for which the entire main is proportioned.

From the foregoing assumptions all parts of the two systems can be calculated. The next step will be to determine the resistance in the first branch of scheme "A" and the last branch of "B," to which should be added the velocity head of discharge and the estimated loss in the breeching, the sum of these items giving the loss for the branch, or the total pressure at the junction. The pressure losses are then figured for each main, up to the last junction of "A" and the first junction of "B." These losses should include the friction as well as an allowance for lost motion in the reducing pieces. The net pressures resulting at the several junctions up to the other end of the main give the balances available for all of the branches, and determines their size and design. The variations of back pressure should correspond to the differences between the junctions. If these differences are too great to be made up by the natural graduation of throat and branch sizes, it is necessary to resort either to throttling or to elongation of branches in order to make up extra resistance for equalization. The latter means is resorted to in the two examples. The branches are made long enough, not only to provide the means for varying back-pressure, but also to give sufficient length for taper pieces that would permit the changes of velocity without material losses of motion.

The two schedules give the complete calculation for the extreme end branches. It is not necessary to go into the same detail for the intervening points of discharge when a system is uniform, as in the present example, as the characteristics and the sizes for the others are intermediate and can be found by interpolation.

A closer analysis of the figures will show that the friction and resistance by elbows make up only a portion of the back-pressure, and are a small part of the total pressure at the junctions. The losses of velocity head, due to the necessarily imperfect form of round pipe breechings, are considerable. They have been estimated to vary from 2/3 down to 1/3 of P_v, according to the taper on the main. The end branches facing the current are assumed to utilize more of the momentum of the flow in the main. The velocity head of discharge P_d is seen to be the largest item making up the total pressure. The pressure drop in the main includes, beside the friction at the stated uniform rate, a loss averaging

about one-third of the reduction of velocity head in each taper piece. Thus the drop in speed from a 25-in. pipe to a 22-in. pipe in scheme "B" for volumes of 124.5 and 89.5 cu. ft. will be from 36.5 to 34 ft. per second, reducing the dynamic head from 1.55 to 1.30 lb. per square foot, as per chart. This difference is not altogether lost, but is partially reconverted into pressure head, same as would be the case in an enlarging cone or nozzle. About one-third of this difference or .08 lb. should be added to the friction head for 15 ft. length which is .225 lb. to find the pressure drop along the main, between the two junctions.

The static head P_s and the resultant P_{v_1} have been figured in order to check P_{v_2}, v_2 the throat velocity and the angle. For one row of branches, at 45 degrees, it will be seen that P_v is practically equal to P_{v_1}, showing the angle to be true. For the smaller branches the uniform angle of 30 degrees is at least approximately correct. An attempt at more accurate sizing, or variations of angle, would not pay in practice.

The pressure losses for the balance of each system are made up of exactly the same items. The totals show a marked difference in favor of scheme "A," for which the pressure and blast power is less than one-half. Conforming to the lower velocities throughout this system it is advisable to keep down also the tip speed of the fan. A wheel of the same diameter as used for B would have to be much wider, or run beyond its best speed to make up for the lack of blast area. A wider blower would also mean lower efficiency, unless two inlets can be arranged. The peripheral speed is taken to be about the same as the theoretical velocity corresponding to the total pressure P. It is read from the auxiliary chart at the intersection of the line of total pressure, with the line for 1 sq. ft. blast area, any abscissa giving the corresponding speed in feet per second, which is the same figure as for the volume in cubic feet per second.

The two examples are also instructive in the showing they would make under reduced head or throttled discharge, which can readily be estimated. A variation of volume carried in any part of main will make a decidedly greater difference from junction to junction in the case of the small pipe, and affect the distribution correspondingly.

CHAPTER X

FORCED VENTILATION

Chart IX.—This chart is intended for the calculation of mechanical ventilating apparatus for buildings. The friction head is computed for square sheet metal conduits of a given cross-sectional area, stated in square inches, with the square feet in round numbers only, shown in dash lines. The losses of head are again figured for an air temperature of 70° F. It should be noted, that the friction head is charted for a length of 100 ft. of straight conduit, while that for air blast, where the straight runs figure relatively more, is expressed for a length of 10 ft. only. The coefficients for resistance by obstruction are given for the features peculiar to ventilating practice for buildings. The total pressure, power, blast area, and theoretical velocity are charted separately, in the same manner as for air blast. The corrections to the volume for leakage in transit, considering the lower pressures, are stated to be somewhat smaller. Other frequent corrections are given for deformed and poorly built sheet iron ducts, for masonry ducts, except glazed tiling, and for duct work of different shapes. Corrections to the total pressure for higher or lower temperature of the air to be moved are the same as for air blast. For variations of temperature within the conduit the data given on the chart for gravity hot air heating may be used.

Outline of the Problem.—The carrying capacity of a given conduit system for ventilation is calculated on the same principles as laid down in the previous chapter for air blast. The problems of distribution are similar, the air being discharged usually against atmospheric pressure. In most instances the velocities through conduits, and especially at outlets, are to be kept within certain limits for various practical considerations, and the total pressure required for moving the air is, to an extent, governed by these. The main problem consists in the equalization of the resistances for the desired volumes. This is accomplished by modifications of duct sizes at such points where the velocities can be varied and by the correct shaping of junctions and other features to con-

form to the natural flow, thereby eliminating as far as possible the uncertain elements. The general rules will apply equally to pressure and to suction.

Heat will sometimes enter as a factor in calculating the available head. A study of such cases is presented in the chapter on gravity movements. The questions of power are generally subordinate to other considerations, but it must be gone into in order to determine the requirements. Incidentally, it will lead to discovering such opportunities for economy, as will always exist.

When all factors entering the problem are duly taken into account, the system equalized to a fair degree and the power established, the desired delivery is assured beforehand without the necessity of throttling excess volume at some points and shunting them to others. Subsequent adjustment of the distribution, especially in unfinished buildings, is vexatious and uncertain. Atmospheric influences constantly change during the process, and hardly ever affect all discharges alike. For these and other reasons it is difficult to effect really fair distribution by dampers and deflectors, and the result is too often a bluff. At best, it involves loss of time, waste of power and presupposes a margin of safety or an excess of capacity nearly always representing extra expense of installation. Moreover, adjustments are most liable to be disturbed by incompetent hands, while a duct system without dampers is not easily tampered with, and not likely to be, if correctly designed.

Computation on such lines involves a thorough study of the factors entering into play and induces intelligent design of conduits, and of apparatus in general, with a view to meeting a given requirement most economically. Again, the application of scientific method results in such proportions and dimensions of conduit, as will minimize the effect on distribution of occasional and partial shutting of outlets.

Atmospheric conditions will affect air distribution more directly than is the case with water or steam, in so far as the points of delivery are influenced by wind action and temperature. While such disturbing factors bear equally on well balanced and indifferently designed systems, the weaknesses of the latter are more liable to show and to be felt under adverse conditions.

Application of Chart IX.—A system of conduits should be considered as a unit, all parts interdepending. It should be pre-

CHART IX

FORCED VENTILATION

THROUGH SQUARE SHEET METAL CONDUITS

At a Mean Temperature of 70° F.

Pressure in lb. per sq.ft. to create velocity $p_v = .075\frac{v^2}{2g}$. $v = \frac{Q}{a}$.

" " " overcome friction $p_f = .075 \times .00624\frac{v^{1.9}}{2g} l \left(\frac{c}{a}\right)^{1.18}$

" " " " resistance of obstruction $p_r = .075 \times 1.25 r\frac{v^{1.9}}{2g}$.

Total pressure required $P = p_v + P_f + P_r$ (net between fan inlet and outlet).

Power to move Q against P in h.p. $\doteq \frac{QP}{550} = \frac{1}{2}$ motive power (approximate).

Theoretical velocity to create P in ft. p.s. $= V = \sqrt{2g\frac{P}{.075}}$ (approximate tip speed of ordinary centrifugal wheel).

Blast area of fan or theoretical outlet area in sq.ft. $A = \frac{Q}{V}$.

Diameter of fan wheel in ft. $D = \frac{60V}{n\pi}$ (approximate).

Revolution per minute $n = \frac{60V}{D\pi}$ (approximate).

Corrections

For leakage add to the volume from 5% to 10% per 100 lin.ft. according to pressure, character of conduits, and ratio of $\frac{c}{a}$.

For deformed and poorly built sheet iron ducts, add from 10% to 30% to friction head.

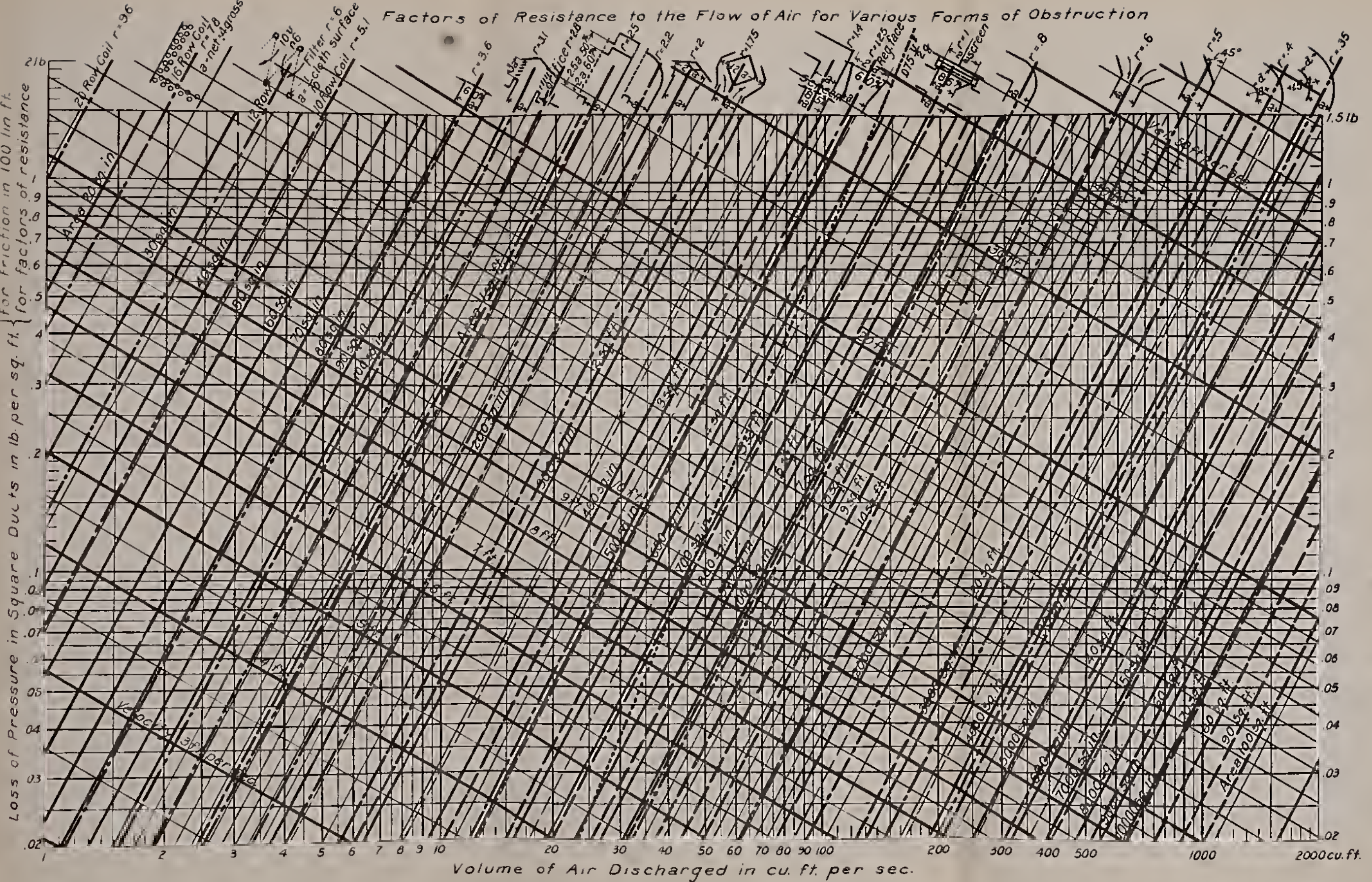
For conduits built of masonry, except glazed tiling, add from 10% to 30% to friction head according to smoothness.

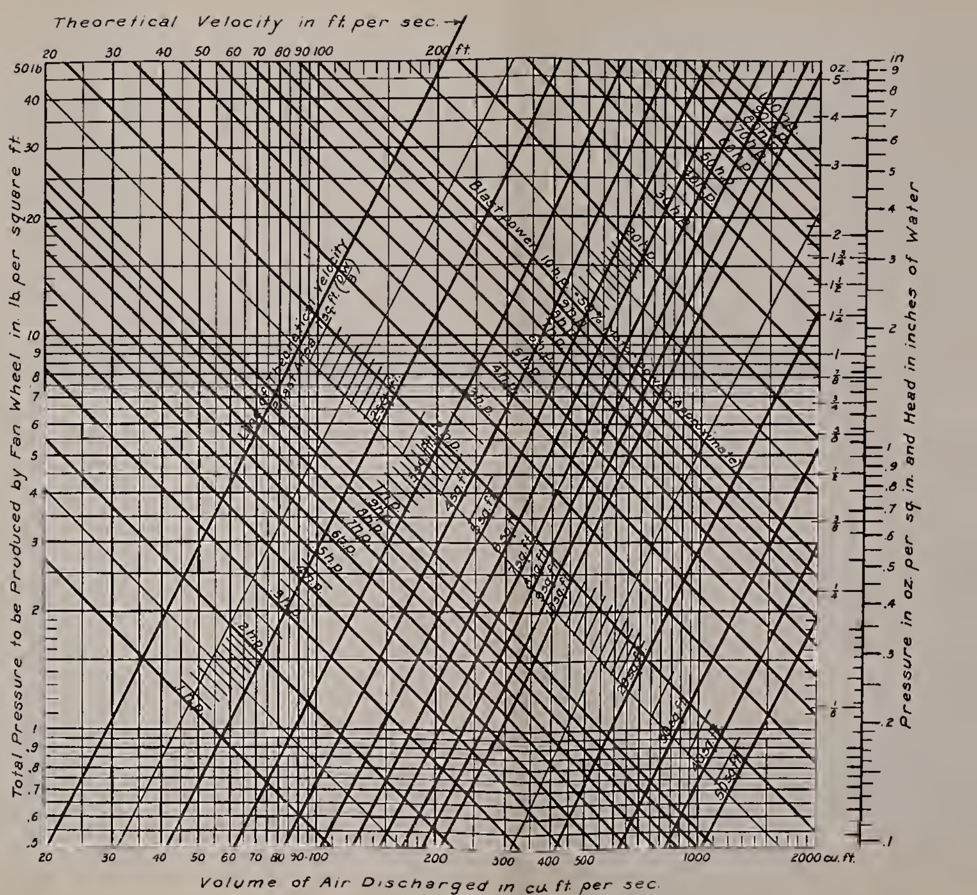
For mean air temperatures in conduits below or above 70° F., correct pressure and horse-power by adding or subtracting 1% for every five degrees of variation.

For round pipes of equal area multiply the friction head by $\left(\frac{d\pi}{c}\right)^{1.18} = .87$.

For flat, rectangular, and odd shapes of cross-section with equal area, multiply the friction head by $\left(\frac{c}{4\sqrt{a}}\right)^{1.18}$.

Factors for rectangular shapes $\begin{cases} 1\times2=1.07 \\ 1\times3=1.18 \\ 1\times4=1.30 \\ 1\times5=1.43 \end{cases}$





sented by a schedule giving at a glance a general survey of the problem in distribution or delivery and providing a convenient vehicle for establishing the various factors, that is, the volumes to be carried at all points, the length of runs to scale, and such sizes as are predetermined by discharge velocities or other conditions. A schedule is useful also for the preliminary sizing of those parts of a system which may be varied for equalization. It is most easily made up from a rough tracing or print of the plan of conduits which will give the horizontal measurements. The vertical runs are of uniform length as a rule for each story and may be stated, together with offsets and other features, for each of the flues. Particulars in design or items of local resistance which do not appear on the plan, can be noted. The figures for the volumes to be carried are to be added up successively, giving the quantity for each part run between junctions with due allowance for leakage. These figures may be distinguished by colors or underscoring. The area of all portions with a given velocity can be read directly from the chart and the corresponding dimensions marked, according to the shape desired.

The sizes of mains, as for air blast, if based on a moderate rate of friction loss will be found to require little revision in equalizing the system. Assuming an initial velocity near the fan outlet as experience may dictate, the chart gives at once the size required for the total volume. The proportionate area for the volumes carried in all other parts of the main is then found directly along the same horizontal line of pressure loss assumed, or established, by the initial velocity. Thus, for instance, a volume of 1000 cu. ft. per sec., moving through a square duct of 34 sq. ft. area, at 30 ft. velocity, will cause a friction loss of .3 lb. per 100 ft. of length. One-tenth of this volume, 100 cu. ft., at the same rate of loss requires 5.8 sq. ft. area at a velocity of 17.3 ft. per sec., and 10 cu. ft. require 1 sq. ft. area at 10 ft. velocity. According to the length of conduit, the power available, and the space conditions, a higher or lower rate of friction loss may be maintained approximately for the entire length of mains. Ducts of decided flat rectangular shape should be sized with allowance for the increased friction head, as stated by the factors for correction. But it should be remembered that these factors apply to the pressure, while the area is to be corrected by the square root of the factor for friction head. Thus a duct 1×4 should have only $\sqrt{1.3} = 1.14$ times the area of a square conduit

for an equal rate of friction loss. As a rule it is proper merely to round up the figures obtained for flat shapes.

The schedule thus prepared now permits the calculation of the pressure losses for equalizing the resistance in connections and branches to individual outlets, and for determining the total pressure and power requirements. The procedure is the same as outlined for air blast, the successive losses being added up, starting at the far end of the line, or at the atmospheric pressure, and noting the increased pressure at each junction up to the fan. The loss for a part run of uniform area made up of several items of friction and local resistance, is rapidly found along the same velocity line, and changes of dynamic head, as far as they involve losses, are found by simple subtraction of the velocity heads for the speeds in question, or parts of the difference as may be estimated from the shape and angle of contraction or enlargement.

For equalizing the delivery through branches, the connections between main ducts and the flues are, as a rule, the parts on which the resistance can most conveniently be varied to suit the need. In some cases it will be found advisable to modify velocities in the flues, but this can be done only to a limited extent. When the losses of head for the outlets and flues at fixed speed are determined, and the pressure maintaining at each junction is established, as stated, the problem of equalization is reduced to one or two tentative assumptions of the size and shape of these connections, which will make up the same total loss for each point of delivery. In selecting the form of resistance to be introduced, it should be remembered that the requisite pressure drop can be effected in three ways, commonly distinguished as friction, obstruction, and deliberate loss of motion. The static and dynamic conditions in the main, also the friction and other causes of back-pressure on branches should decide the method to be adopted in each instance, as previously outlined in the chapter on junctions.

Friction should be depended upon wherever practicable. This is done by keeping up the full velocity obtainable, or natural to the branch at the angle chosen, as far as necessary to use up the excess of pressure. To deter the flow by extra obstruction as, for instance, by using close elbows instead of bends, may answer well enough sometimes, but the method is not conducive to accurate work. Conversion of static into dynamic head, to be

deliberately lost by discharge through an orifice, is necessary where branches are too short to cause a desired friction head. The velocity obtainable through diaphragms or orifices is readily found on the chart, if the back-pressure is known, but the extent to which the velocity head of discharge is lost by impact and eddies, or is reconverted into pressure by the throat, is difficult to estimate. Hence it is best to depend upon friction head to equalize resistances in uneven length of branches and for decreasing pressure at junctions. This simply means larger areas for greater length, or reduced area on short branches, for which the friction is easily and accurately calculated. It saves space and material, and results in lower velocities than those obtained by orifice, or half closed dampers, which are liable to cause noise.

The right hand portion of the chart for forced ventilation gives the theoretical blast power and blast area for any volume, to be moved against any pressure. It contains also two separate scales for converting the pressure from pounds per square foot into inches of water column and ounces per square inch. The line for 1 sq. ft. blast area, at its intersections with the abscissæ for pressure indicates also the theoretical velocity for a given total pressure, the figures in feet per second being identical as those for volume.

The peripheral speed of the fan is thus practically read from this chart, subject to certain modifications. The blast or equivalent area corresponding to the total pressure determines the size of the fan wheel for a given type and proportion. For the ordinary six or eight blade paddle wheel the blast area A equals about $\frac{DW}{3}$, W being the width at circumference. $\frac{DW}{4}$ is approximately true if W is the full width at inlet, or that of the housing, as given in the catalogues.

As previously stated, the actual motive power will depend upon the efficiency of the fan, which depends again upon the nearness by which the commercial article will fit the case. The methodical use of the charts, by establishing the true requirements as to static and dynamic pressures, permits the selection of the type of wheel, with dimensions and angles of discharge that will at least approximately meet the theoretical lines drawn in the general chapter. The highest efficiency for a given machine will then coincide about with its working conditions and assure economical operation.

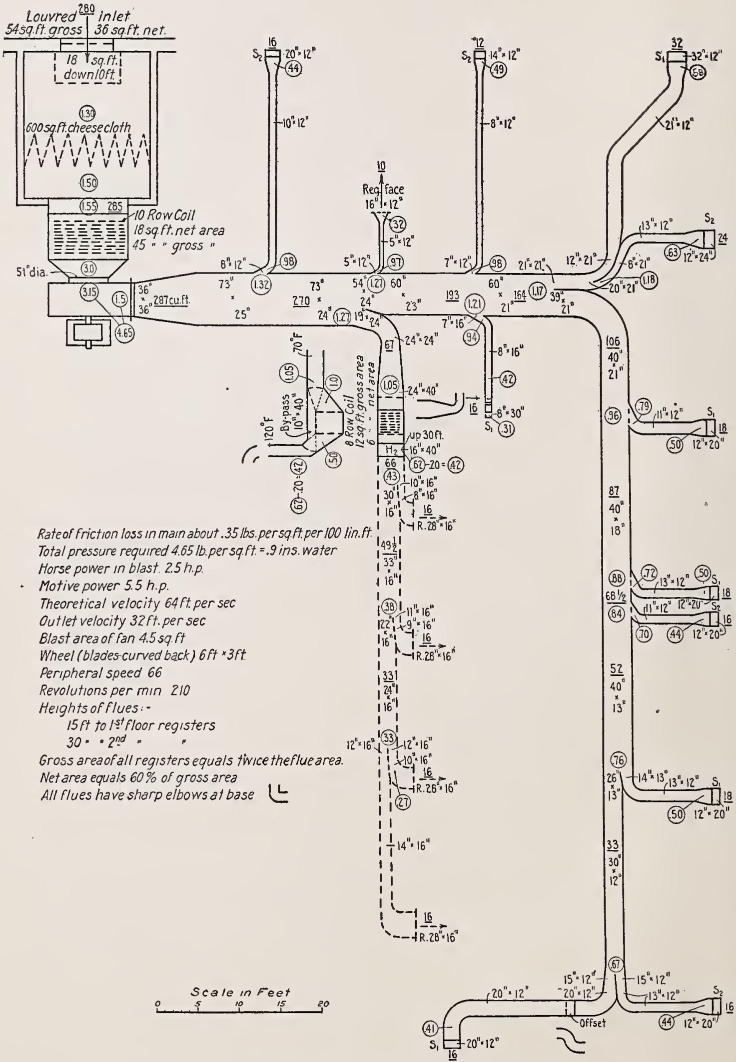


FIG. 30.—Example of duct system for ventilating.

Example of Forced Ventilation.—Fig. 30 represents a typical duct system for a ventilating apparatus. The scale plan of the ductwork is used as a schedule for calculation. The figures for the volume of air per second for each flue are underscored. The pressures are noted in circles. The rate of friction in this case has been assumed to average about .35 lb. per sq. ft. for 100 ft. length, including the correction for flat shape. This gives as much difference in pressure between the extreme ends as can be conveniently equalized by friction in the connections to flues. A lower rate of loss would require larger ducts, while a greater drop would involve a higher initial pressure, and make it more difficult to equalize, the first branches becoming still smaller.

The sizes of the registers and flues being determined by various practical considerations, the pressure losses down to the base of each flue are figured definitely and marked. A branch of the full size of the flue at the extreme far end of the line gives the loss of pressure to the first junction. This loss includes a portion of the decrease in velocity head from the main to that branch, the former being sized to conform to the .35 lb. rule for the main. The taper piece at the breeching is designed to reduce this item to a small, almost negligible fraction. It will be seen that breechings for runs of uneven length taper unequally, but converge at equal velocity. Thus, the second or short branch is contracted beyond the junction in order to bring its resistance up to that of the first named long one.

From the breeching at the far end, up to the fan outlet, the pressure losses between junctions are added up successively. They include slight allowances for the decrease in the velocity head opposite the branches or for the taper pieces on the main, where changes in speed take place. The branches are mostly taken from a flush side of the main. The sizes of connections from the junctions to the bases of the flues are chosen to offer a resistance equal to the difference in pressure between the two points, including the probable loss due to imperfection of breechings. The velocities in these connections vary from that of the main to about 25 per cent. above it. The angles of branch pieces are increased correspondingly toward the fan, conforming to the higher speed of outflow necessary for equalization. The loss by eddies and impact is thereby reduced to a small fraction of P_v.

The head produced by the supplementary heating coil is available in winter only. Without the heated air column, the discharge would be appreciably reduced. A by-pass around the coil for use in summer is calculated to offer a resistance equal to the obstruction of the coil, less the buoyancy of the warm air in winter, which figures .2 lb. per sq. ft. for 50 degrees difference and 30 ft. height as obtained from the chart for gravity hot air heating.

The main duct at the fan outlet is designed with an easy taper in order to save loss of motion in getting down to the initial velocity desired. To enlarge the fan outlet to the full size of the main would not avoid this loss, since the same reduction in speed would then simply take place inside of the housing, possibly under less favorable conditions and probably without being taken into account. It is better to reduce the outlet of the ordinary blower down to the equivalent area, if a gradual enlargement of the duct can be secured, such as will convert the excess of dynamic head into static pressure. For the present example we find the theoretical speed for 1.5 lb. pressure at the outlet to be 36 ft. per sec. and the equivalent area 8 sq. ft. The regular outlet would be 10.5 sq. ft. The actual area chosen is 9 sq. ft. The main duct is 12.6 sq. ft., but the reduction in speed is so gradual that practically no loss occurs from that cause. The drop of .18 lb. from the nearest junction is made up mainly of friction and the resistance of a slight bend. The pressure drop for the tempering coil and casing are stated as separate items, also that for the filter and the air intake.

The total pressure required gives a tolerable speed for the fan, indicating blades with a backward curve to bring the speed of exit from the wheel nearer to that of the outlet, which would reduce the losses of motion and tend toward higher efficiency. The peripheral speed is taken slightly in excess of the theoretical, to make up for the extra resistance presented by the single inlet. The latter also involves slightly higher motive power, which is based in this case on somewhat less than 50 per cent. efficiency. It is probable that this gives a safe margin of capacity, as the fan efficiency, under proper selection, is likely to be higher. It would be easy, in this case, if desired, to bring the motive power within 5 h. p. by reducing the resistance through the coils or intake and make it safe to use that size of motor.

Example of Coupling Fans.—Occasionally it is desirable to

couple together a blower and an exhauster. Assuming the duct system for each to call for different volume and pressure, to be created by the same motor on the same shaft, the problem will be to find the proper relation in the diameter and width of fan wheels, so that each may work to best advantage.

Taking for example an air supply system, to deliver 300 cu. ft. of air per second, against a pressure of 3/4 oz. or 6.75 lb. per sq. ft. The blast area is found to be near to 4 sq. ft. The theoretical velocity is 77 ft. per second. The diameter of a fan wheel with standard proportions, for which the width over all, $W=0.5D$, and $A=\frac{DW}{4}$, will be $D=\sqrt{8A}$, in this case 5.66 ft. For the nearest commercial diameter, 5 1/2 ft., the width over all should be $W=\frac{4A}{D}=2.91$ ft. or, say, 3 ft. The speed for that diameter is $\frac{60V}{D\pi}=267$ rev. p. m.

Coupled to the same shaft, the other fan must necessarily make the same number of revolutions, at which it is to deliver, for instance, 250 cu. ft. per sec. against a pressure of .55 oz. or 5 lb. per sq. ft. According to the chart, this requires a blast area of 3.8 sq. ft. In order to create the desired pressure this wheel is to have a peripheral speed of 66 ft. per second. Its diameter is, therefore, predetermined, and will figure $D=\frac{60\times 66}{267\times 3.14}=4.7$ ft. and for the same relation of $\frac{DW}{4}$, the width will be $W=\frac{4\times 3.8}{4.7}=$ 3.2 ft. The question now arises which is the proper commercial size to select. The next larger diameter will give more, and the nearest below an appreciably lower pressure than that needed.

If the main ducts leading to this fan can be increased and divided for two inlets, and the resistance eased sufficiently thereby to bring the diameter area to the regular size of 4.5 ft., making $A=4$ sq. ft. and $W=\frac{4\times 4}{4.5}=3.55$ ft., then the next smaller diameter will prove to be more advantageous, since both the first cost and operating expense are kept down.

CHAPTER XI

HOT AIR HEATING AND VENTILATING BY GRAVITY

Chart X.—The movement of air created by difference in temperature, as utilized for warm air heating and ventilating by gravity, takes place at moderate velocities. The chart is accordingly made to cover a lower range than that selected for VIII. and IX. The losses of head by friction are again charted for lengths of 100 ft. for square sheet metal ducts. The area is given in square feet and fractions thereof, so that the three charts should supplement each other to an extent, one giving the diameter of round pipes, the other the area in square inches and the third the area in square feet. The features of obstruction are partly the same as those of the other charts, with some additional items referring mainly to gravity heating.

The corrections for different kinds of conduits and for odd shapes are the same as those for forced ventilation. No allowances need be made for leakage, except when dealing with unusual pressures or with very leaky ductwork. The corrections for temperature as affecting friction and head available are stated on the chart and further explained in the text.

The auxiliary diagram gives the total pressure due to differences of temperature above 70° F. in heat flues, and below 70° F. between room or vent flue and the outside air, for any height, with the idea of determining the theoretical velocity, from which the actual flow can be estimated roughly and assumed as a basis for final calculation. The initial temperature of 70° F. covers the normal conditions. For abnormal room or vent flue temperature the factors for correction are stated.

Aero-motive Force for Gravity Circulation.—The force creating what may be called a natural flow of air is the difference in weight between two connecting unbalanced air columns under the same atmospheric pressure. It can be expressed as the product of the mean excess of density of one side over the other, and the height for which this excess maintains. If w_1 and w_2 are the

CHART X

HOT AIR HEATING AND VENTILATING BY GRAVITY

THROUGH SQUARE SHEET METAL CONDUITS

Total pressure created in lb. per sq.ft. $P = .075\left(1 - \frac{530}{T_f}\right)h$ — for heat flues above 70° F.

Total pressure created in lb. per sq. ft. $P = .075\left(\frac{530}{T_o} - 1\right)h$ — for vent flues at 70° F.

T_f = absolute flue temperature, T_o = absolute outside temperature in degrees F., h = actual height of draught.

For assumption of correct temperature and height see text.

For vent flues at higher temperatures figure margin above 70° F. as heat flue, adding the pressure to that due to margin below 70° F.

Pressure in lb. per sq.ft. to overcome friction $p_f = .075 \times .00624 \frac{v^{1.9}}{2g} l \left(\frac{c}{a}\right)^{1.18}$

" " " " resistance of obstruction $p_r = .075 \times 1.25 r \frac{v^{1.9}}{2g}$.

" " " create velocity $p_v = P - (P_f + P_r)$.

Actual velocity obtainable $v = \sqrt{2g \frac{p_v}{.075}} = \frac{Q}{a}$. Q = cu.ft. per sec. at 70° F.

Theoretical velocity $V = \sqrt{2g \frac{P}{.075}}$.

Corrections

The pressure created is subject to correction only when the room or vent flue temperature is above or below 70° F.

For T = 510° F. = room or vent temperature of	50° F., multiply P by	1.08
" = 520° " " "	60° " "	1.04
" = 530° " " "	70° " "	1.00
" = 540° " " "	80° " "	.96
" = 550° " " "	90° " "	.92
" = 560° " " "	100° " "	.89

The pressure to overcome friction and resistance in conduits is to be corrected for temperature in ducts and flues, etc., multiplying the charted pressure losses by the factors below.

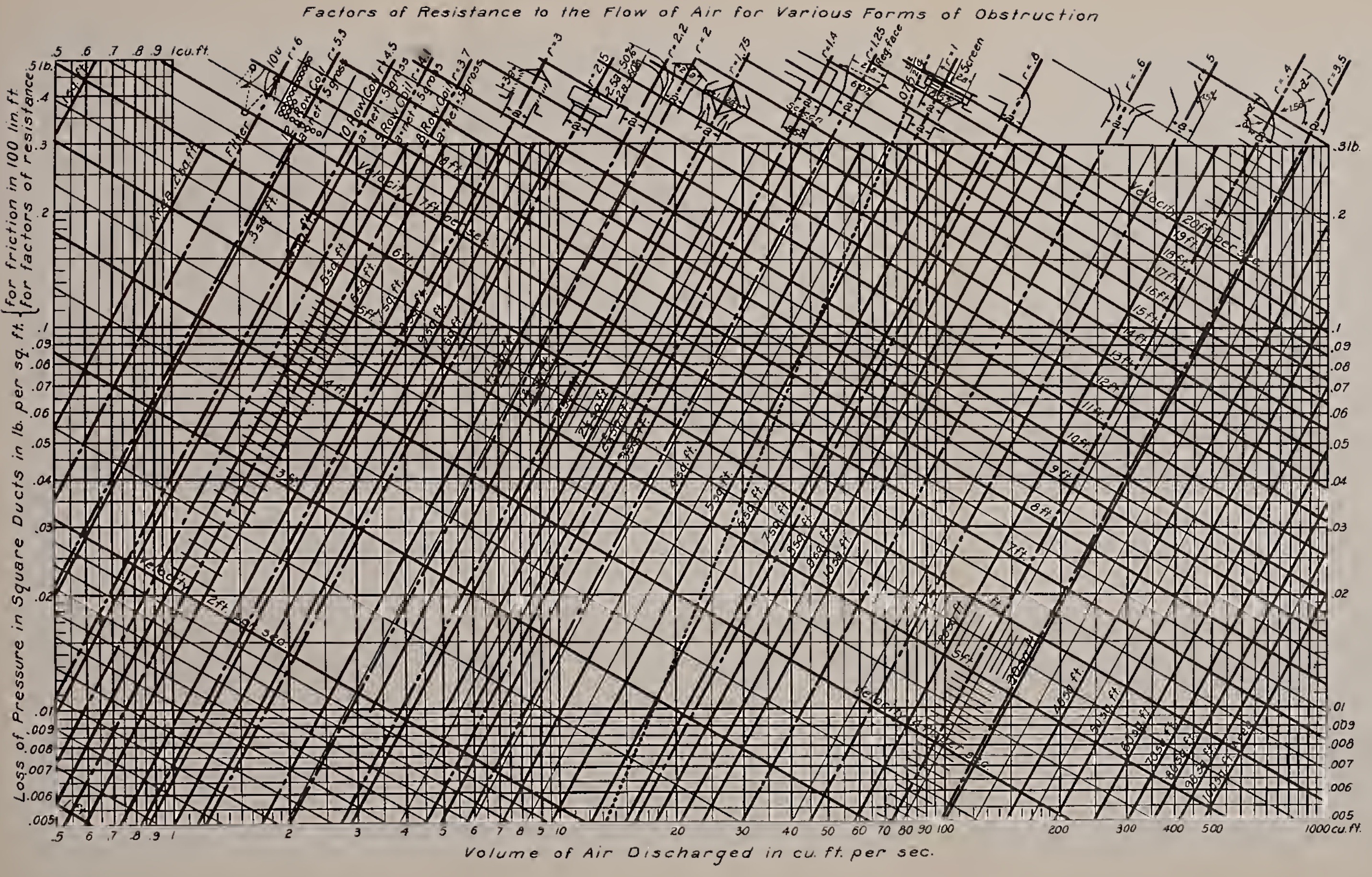
For t_f = 0° F. – .87	For t_f = 70° F. – 1.00	For t_f = 140° F. – 1.13
" = 10° F. – .89	" = 80° F. – 1.02	" = 150° F. – 1.15
" = 20° F. – .90	" = 90° F. – 1.04	" = 160° F. – 1.17
" = 30° F. – .92	" = 100° F. – 1.06	" = 170° F. – 1.19
" = 40° F. – .94	" = 110° F. – 1.08	" = 180° F. – 1.21
" = 50° F. – .96	" = 120° F. – 1.10	" = 190° F. – 1.23
" = 60° F. – .98	" = 130° F. – 1.11	" = 200° F. – 1.25

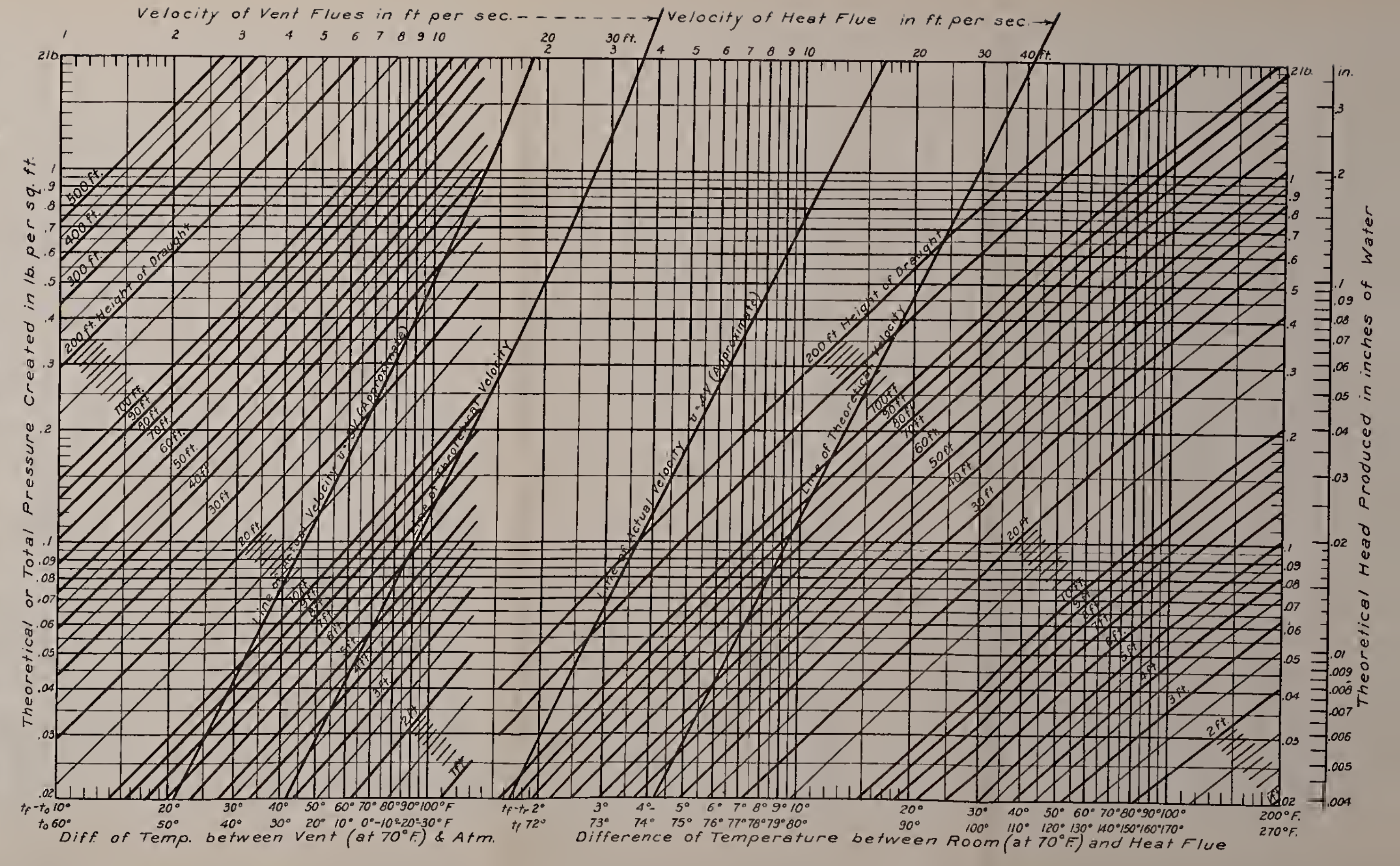
For conduits of masonry, except glazed tiling, add from 10% to 30% to the losses by friction, according to smoothness.

For round pipes of equal area multiply by $\left(\frac{d\pi}{c}\right)^{1.18} = .87$.

For flat, rectangular, and odd shapes of cross-section with equal area, multiply the friction head by $\left(\frac{c}{4\sqrt{a}}\right)^{1.18}$

Factors for rectangular shapes: $1 \times 2 = 1.07$; $1 \times 3 = 1.18$; $1 \times 4 = 1.30$; $1 \times 5 = 1.43$





respective weights per cubic foot and h the actual height in question, the resulting pressure will figure $P = (w_1 - w_2)h$.

The height which would give the theoretical velocity of fall is not identical with h, but equal to the imaginary difference H which the two unbalanced air columns would show after reassuming the same temperature. (See Fig. 31).

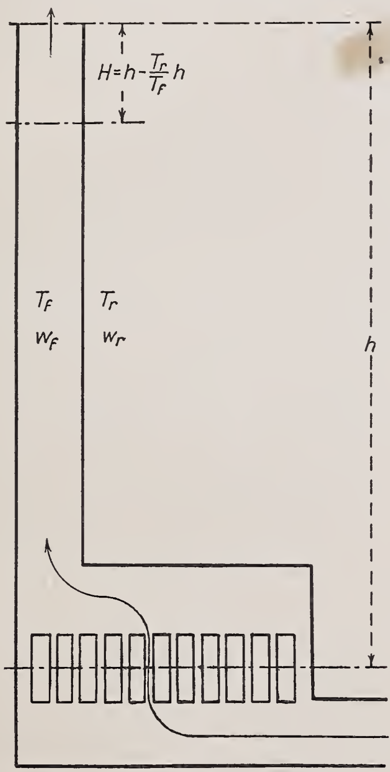


FIG. 31.—Effective head of heat flue.

This differential height really represents the expansion of the heated column. It is the head H available for creating the flow. If the velocity and area are given, it is the head required and to be created by temperature difference or height. Rietschel expresses it by the formula $H = \left(\frac{1}{1 + \alpha t_1} - \frac{1}{1 + \alpha t_2}\right)h$.

In this expression t_1 and t_2 are the respective temperatures above the freezing point, and α the coefficient of expansion, also

at 32° F., at which the density $w = .0807$. The product wH must then equal the pressure P as defined above.

If we substitute the room temperature, as the initial one, which is pretty uniformly at 70° F., the formula may be simplified to read

$$H = h - \frac{T_r}{T_f}h = \left(1 - \frac{T_r}{T_f}\right)h$$

wherein T_r and T_f are the absolute temperatures in room and flue. The resulting pressure in this case is

$$P = w_r H = w_r\left(1 - \frac{T_r}{T_f}\right)h = (w_r - w_f)h$$

Inserting the value of T_r for 70° F., we have

$$P = 0.075\left(1 - \frac{530}{T_f}\right)h$$

The pressures and theoretical velocities for heat flues are computed and charted on this basis.

For vents we can write correspondingly

$$H = h - \frac{T_o}{T_f}h = \left(1 - \frac{T_o}{T_f}\right)h$$

and $$P = w_o\left(1 - \frac{T_o}{T_f}\right)h,$$

which forms will give the effective height and pressure measured by an air column of outside temperature T_o. Conforming to all other tables, in which the heights and resulting velocities are given for air at 70° F., we may also write

$$P = w_f\left(\frac{T_f}{T_o} - 1\right)h \text{ which must equal } (w_o - w_f)h.$$

As a rule, T_f is both the room and flue temperature and in that case $P = .075\left(\frac{530}{T_o} - 1\right)h$

The pressures created have been calculated by this formula, but the theoretical velocity is figured by the first one, giving the true effective height H which governs the fall.

As indicated in Fig. 32, the height of the columns in question is not identical with that of the flue, but should be figured from the mean level of leakage into the room, at which level it is assumed that the air is being heated. This may not always be

strictly true, but is probably the fairest average assumption where the inward leakage is not influenced by suction or pressure on the room, as will be explained in the chapter on neutral zones.

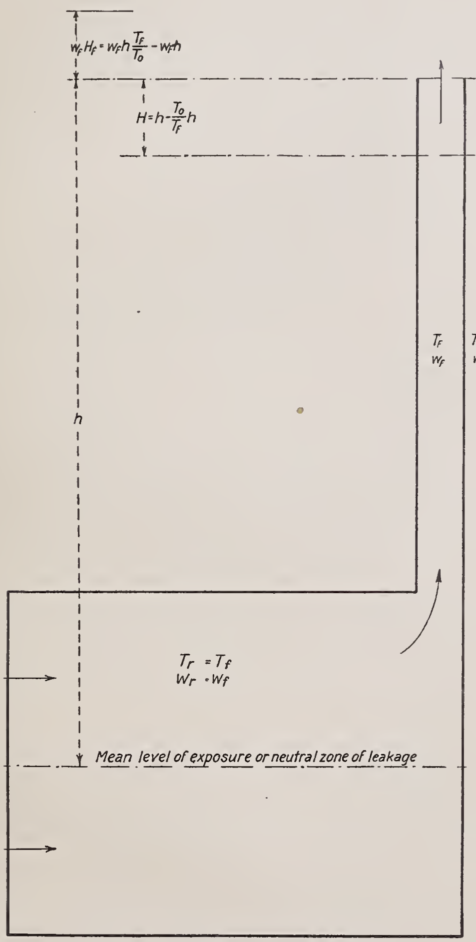


FIG. 32.—Effective head of vent flue.

The correction to the pressure for any room or vent flue temperatures above or below the normal 70° F., is figured simply by the ratio of P_c to P wherein

$$P_c = w \frac{T_r}{T_{rc}} \left(1 - \frac{T_{rc}}{T_{fc}}\right) h$$

The factors are given on the chart for a range from 50° F. to 100° F. The corresponding effective height H_c, and the resulting theoretical velocity V_c do not vary at the same ratio, but the error in the actual velocity is of no consequence for approximate sizing and does not bear on the final result at all, if the velocity obtainable is verified by the final calculation of pressure losses.

For accurate calculation it is essential to correct the friction, resistance and velocity heads according to the temperature maintaining in the various parts of conduit. To facilitate this, factors are given for a liberal range, expressing the ratio for the increased losses due to greater velocity under expanded volume, and allowing for variation in density, according to the formula

$$P_c = \left(\frac{T_f}{530}\right)^2 \times \frac{w_f}{0.075} P = \frac{T_f}{530} P$$

If the pressure loss is read on the basis of the changed volume, the corrected pressure is

$$P_c = \frac{w_f}{0.075} P = \frac{530}{T_f} P$$

The velocity head, or the pressure expended in motion is the balance left, or the net result after deducting from the total the losses by friction and local resistances

$$P_v = P - (P_f - P_r)$$

Since P_f and P_r depend upon the actual velocity v, which is the unknown quantity, the latter must be found by tentative assumption, until the equation between the velocity obtainable and that required is complied with, that is, until

$$v = \sqrt{2g \frac{P_v}{w}} = \frac{Q}{a}, \text{ wherein } Q = \text{ cu. ft. per sec. at } 70° \text{ F.}$$

If Q is corrected for flue temperature $v = \frac{T_f}{530} \times \frac{Q}{a}$

Outline of the Problem.—Warm air heating by the indirect medium of hot water or steam is usually designed with individual heating stacks for each flue or room, connected with one or more air intakes, sometimes with filters and tempering coils. We have, therefore, the problem of equalizing the delivery to several flues with unequal resistance and unequal pressure available, same as is met in hot water heating by gravity, except that not only the levels are different, but usually also the temperature range. The situation is more complex also in regard to establishing the true

effective temperatures, or differences in density, and the heights that come into play. This feature requires study, particularly for open or interrupted circuits, subject to external influences, which are the rule rather than the exception. The greater element of uncertainty in the calculation of heat flues and vents lies in the assumption of these two factors. It seems necessary to define them more clearly.

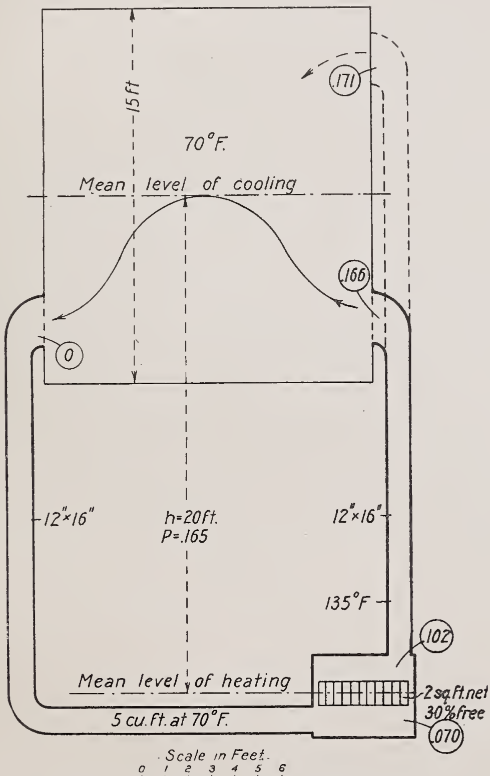


FIG. 33.—Indirect heat with closed circuit.

Mean Levels.—Any closed circuit within which air is alternately cooled off and heated again has two turning points where the fluid begins to rise and to fall. These points are naturally at the mean level at which heat is applied and given up. For hot water heating they are recognized to be the mean levels of boiler and radiators. For indirect warm air heating it is the stack and the heat transmitting surfaces of rooms, as illustrated by Fig. 33.

The vertical distance between them marks the height of draught h, which may be greater or smaller than the height of the flue proper. These mean levels do not necessarily fall together with the two points on a circuit where suction changes into pressure and *vice versa*, but will often fall apart, especially with unequal resistance in the up- and down-take flues and ducts. In a case like Fig. 33, the room would be put under a slight pressure if the heat flue is larger or shorter than the return flue, and under vacuum, if this condition be reversed.

Neutral Zones.—Recknagel in his studies of aerostatics in rooms has termed the strata, at which atmospheric pressure is maintained "neutral zones." On a closed, and tight circuit the points at which there would be neither excess of pressure nor suction, would be defined solely by the friction and resistance along the conduit. Such a condition, however, never exists in practice. Leakage at points where pressure or vacuum prevails will often depress or raise the neutral zones very considerably. On open or interrupted circuits, as for instance heating arrangements with intakes and vents to and from the atmosphere, external wind pressure is liable to change the zone. Also the effect of one flue over another may radically disturb the static pressures along the line, even to the point of reversing the flow.

While the shifting of the neutral strata will have no bearing on the volume of air in transit on a closed circuit, since one side can make up for the other, the raising or lowering of the zones on an open system virtually has the effect of increasing or decreasing the height of draught by adding to, or taking from the head available for the movement in a flue. A too large heat flue, for instance, will depress the neutral zone in the room and decrease the height of draught necessary to create a given movement of air by creating an excess of pressure. The effective height of the vent is thereby correspondingly increased and its size may be smaller. The same effect may be due to wind pressure on the air intake, or suction on the vent.

The influence of leakage on the neutral zones or the height of draught can be estimated with a fair degree of assurance and should be taken into consideration, not only in its possible bearing on the volume of air or the amount of heat required, but as affecting the static pressure in the room, and thereby the ingress and egress through heat and vent flues. A room, or building as a whole, kept at higher temperature than the atmosphere, but

communicating with it by leakage, will form an unbalanced column of lighter air tending to escape on the top and drawing in heavier air at the bottom. As illustrated by Fig. 34 this tend-

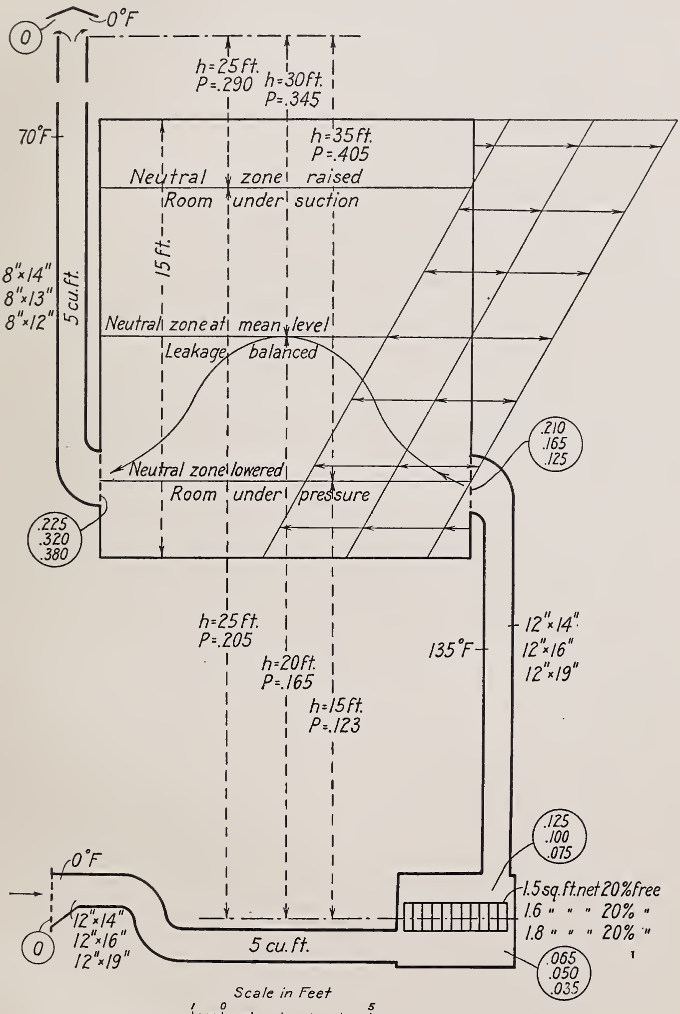


FIG. 34.—Indirect heat and vent open to atmosphere.

ency for ingress and egress increases with the vertical distance from the neutral zone, that is, with the height of a room, or building. It also increases with the difference in temperature between

the inner and outer air. Tall rooms may show considerable pressure and outward leakage in the upper strata, and show a strong tendency to indraught through doors and windows at a lower level. If this latter tendency is to be avoided, the air supply flue must be ample enough to reduce its height of draught or bring down its upper strata to a level below the points of inward leakage. In other words, the neutral zone is to be lowered below the level of windows and doors, and the system calculated and proportioned on the basis of these heights. The extra pressure to be exerted is easily figured as the product $(w_o - w_r)h$ or read from the chart, h being the height for which the zone is to be depressed. This applies to forced ventilation as well, if a room is to be put under fan pressure in order to avoid in-draughts.

In a similar way, the neutral zone may be lowered by back-pressure, caused, for instance, by insufficient vent capacity. Such restriction of the volume at once reduces the effective height of draught for the heat flue. When the room is air-tight and the vent closed, the movement of air in the heat flue ceases because the height of draught is reduced to nothing. The upper and lower neutral zones fall together. While this condition maintains there may be still some heat delivery through up and down circulation within the same flue, but the least additional pressure on the room through inward leakage will reverse the flow.

To what extent a moderate change in pressure, or a shifting of the neutral zone will bear on conduit area, is illustrated on Fig. 33. A difference of 5 ft. is seen to modify the flue sizes, especially that from an indirect stack, for which the margin of pressure to create the motion is materially affected.

The height of a flue itself, as already mentioned, and the difference in level between heating and cooling surfaces are, therefore, by no means identical with the effective height of draught, but, as may appear through further consideration, it is generally desirable to make them so. That is, the neutral zone should be laid at the mean level of cooling, which, at least theoretically, is also the proper level for a warm air discharge.

Correct Height of Draught.—It will be realized that in practice various conditions will modify the height of draught to be taken as a basis for calculation. Living rooms on the lower stories of residences, when open to a stair hall, will show an increased height of draught, which may be estimated from the number of

stories so connected. The mean level of exposure may also affect the height to an extent. Allowance should be made for wind pressure, unless its effect is neutralized by the situation of the air inlet. Rooms with only one side exposed are more liable to be under back-pressure through wind action when the warm air is most needed. If sufficient vent capacity is not provided, it may happen, and it does happen sometimes, that the room air escapes down through the heat flue, carrying the upper neutral zone even below the level of the stack.

The investigation into the true height of draught should lead not only to greater assurance in the calculation of the expected flow, but should also lead to the modification of the height of the zone, placing it approximately at the most advantageous level through the proper disposition of air inlets, vent outlets, the closing of stair wells or providing an escape in the form of a fire-place or vent flue. Other means will suggest themselves tending to keep the movement of the air under control, as the case may require. It is advisable, for example, in fact often necessary, to expose the air inlet to the same winds as the rooms which they supply. Owing to inward leakage under pressure from the outside the heat requirement is greatly increased. Simultaneously the neutral zone is depressed, and the supply of hot air is curtailed, unless advantage is taken of this same wind pressure to over-balance the inward leakage of cold air, by directing it also upon the air intake to the heating stacks. If so arranged, it is proper to figure on the wind pressure for overcoming a portion of the resistance. This may include that presented by inlet screens, air filters and tempering coils, but should not exceed the proportionate allowance made to the hot air supply for the exposure of the room, as will be explained by the example for warm air heating.

The height of draught for natural vents is defined, like that for heat flues, by the vertical distance between the mean levels at which the air is heated and cooled, but modified again by external influences. The lower strata of balanced pressure usually lies within the room where the heat is applied or present in the warmed air contents, but it may be above or below its mean level, if put under pressure or vacuum. The upper strata is clearly at the outlet of the flue into the atmosphere. The same external or internal influences that may raise or depress the neutral zone, and increase or decrease the height of draught for the heat flue will also affect the height of the vent, but inversely. A heat flue

of greater capacity than the vent will put a room under pressure, depress the neutral zone, encourage outward leakage, making the vent more effective by reason of greater height of draught. Wind pressure on the room will have a similar effect. On the other hand, suction through communicating air shafts or stairways will oppose the upward movement in the flue by reason of a raised neutral zone and decreased active height.

The heights of draught for heat and vent flues of the same room are therefore interdependent or complementary when external influences are eliminated, but this is hardly ever true in practice. For that reason it is best not to consider them a single circuit, but to calculate each by itself, on the basis of the proper height. Due consideration of this height in question will lead to the discovery of disturbing influences, which may then be eliminated or counteracted intelligently, as the case may demand. The bearing of this on the results is self evident.

Among the disturbing external influences on vent capacity it is important to consider the action of winds upon the discharge opening. When located on the roof, these outlets are exposed to dynamic forces from all directions, tending to assist or upset the flow. These forces should be neutralized or regulated as much as possible by some form of top which will deflect the outer currents in the direction of discharge. Each case ought to be studied individually, and the right form selected on its merits. It is not proper to depend on these devices for creating suction, since that effect is available only under wind action, that is, when extra movement is least needed. It should also be remembered that vent caps in any form giving free exit, while stopping back-draughts from above, cannot prevent reverse action caused by suction from below, due to some disturbance of the neutral zone within. Such disturbances, aside from spontaneous effects through opening of doors or windows, are frequently due to peculiarities in construction which might have been avoided, or through the action of other flues when based on erroneous assumption of height and temperature.

Temperature as a Factor.—The difference in the mean densities of the rising and falling air columns representing the second factor in the product of height and weight $h\ (w_r - w_f)$, is determined roughly by the temperatures of the room to be warmed and of the hot air flues, or by the temperatures of the flue and of the atmosphere for vents. The true differential weights, which

create the power do not always correspond to these temperatures and are subject to modification.

The temperature of the room air is presumed to be known, and can be taken as a basis for the density w_r of the colder column balancing that in the heat flue, but the temperature in the latter is liable to change in transit. When flues are placed in inside walls and are lined, the drop is a negligible item. Flues built into outside walls should either be insulated against heat loss or allowance should be made in the heating surface to make up for the same by higher initial flue temperature. For a given degree at the register this will raise the mean temperature, but, as a rule, it will not pay to take this into account.

In the case of ordinary natural vents discharging into the atmosphere, the room temperature may vary considerably from the average maintaining within the flue, and the draught is thereby materially affected. In tall rooms for instance, under certain conditions, the temperature near the ceiling is liable to be several degress higher than near the floor. If a top register is used, the calculation of the vent capacity should be based on a higher temperature. This is all the more necessary as the friction head is decreased by a shorter flue, and, although this tends to raise the neutral zone, the decreased height is not likely to make up for increased heat. The temperature of a vent may also be affected by cooling or heating in transit, according to surroundings. Such temperature changes can only be estimated roughly, but it is nevertheless important to consider them, if only to avoid extremes and uncertainties in general, by appropriate location or by special measures. Any heat supplied or lost in the vent should be estimated and the mean temperatures corrected accordingly.

When the air supply to a room is determined solely by its heating requirements, the vent can be calculated for full capacity at the lowest outside temperature. It will then draw less air in warmer weather and regulate the amount of air supplied to an extent, approximately according to need. If a definite volume of air is to be introduced for ventilating purposes, irrespective of heating, its temperature must vary, and with it the pressure of inflow. The vent is then calculated to move an equal volume up to a stated outside temperature that may be varied according to the nature of the case. Provision should be made to control the action of such vents in colder weather. It is generally impracti-

cable to base the capacity of vents on outside temperatures above the freezing-point, since it will result in excessive flue sizes and lead to difficulties in regulation. If such vents are to have full capacity up to oustide temperatures that will permit the opening of windows in a crowded room, or if the flue space is limited, the increased draught power must be secured by heating the vent up to the required differential temperature, or by mechanical means, when indicated.

Distribution.—The problem of distributing hot air, or of collecting vents is essentially the same as outlined for forced ventilation, but complicated somewhat, as stated, by the differentiation of the total pressure for the various points of delivery. When the heat source is central, only the heights will vary, except so far as the effective head may be reduced by heat lost in transit, as in the case of hot water heating, or ordinary furnace work. With an individual heat source for each room the density will also vary, and it becomes necessary to figure the pressures available in each case, as a basis for distribution. A system of ducts should be proportioned accordingly to balance resistances against the pressure created in each flue by its own heat.

The effect on distribution of throttling one or more outlets is less disturbing on the delivery of heat flues in so far as each provides its own motive power. If a main duct is designed for full capacity, the volumes passing through the flues left open is but slightly increased as the friction in the common passages is lessened, that part of the resistance being small compared with the balance, represented by the individual stacks, flues, and registers.

While there is a tendency to self-regulation in a warm air heating system owing to the probable increase of flue temperature at reduced velocity, this tendency is by no means as pronounced and dependable as in the case of hot water circulation, since the heat supply is not directly controlled by the air circulation, and other factors will enter into play.

Under reduced head, that is, with lower flue temperature, owing to a decreased output of the heat sources, the distribution is influenced mainly by the heat delivery to the stacks, which is presumed to be equalized. Since a shortage in heat emission means also a reduction of draught power, volume, and efficiency of heating surface, all interdependent, the effect is difficult to estimate. It becomes noticeable for instance where direct hot

water heating surfaces on upper stories are allowed to get the best of the circulation to the stacks.

Some of the conditions described as controlling the action of gravity heating and ventilating apparatus may appear at first glance to be too elusive to permit a safe estimate of the factors for reasonably close computation. However, the study of these conditions in itself gives the means to gain control over them to a greater extent, or to recognize and meet them as the case may be. The forces for creating natural movement are as positive as any that may be created by mechanical means, but limited as to potential. If the resistances can be suited to the pressure that may be available, a reasonably accurate prediction of the result is assured. The charts make such calculation feasible and practicable.

Application of Chart X.—The method of using the chart for gravity apparatus differs from that used in mechanical ventilating principally in the reversal of the proceeding as to establishing the working pressure. The pressure created, or available, is in this case the governing condition to be established first. When the principal factors, height of draught and temperature, have been sized up the resulting pressure and theoretical velocity can be read directly from the chart. They are subject to correction only when rooms to be heated or ventilated are to be kept at considerably higher or lower temperature than the customary 70° F.

For purposes of estimating, the table of pressures created gives the approximate actual velocity for various conditions. Indirect heating apparatus presents considerable resistance to the flow of air. Ordinarily, the actual velocity will not be more than 40 per cent. of the theoretical. With air filters and tempering coils, it is likely to be even less. In vent flues discharging directly to the atmosphere without main ducts, the actual velocity is generally 50 per cent. of the theoretical, sometimes more.

These approximations should be used as a preliminary assumption for calculating the friction, resistance and the velocity head. The sum of these *must* always equal the pressure available, and *should be made to equal it for the requisite volume of air.*

The volume is supposed to be measured at a temperature of 70° F. It will be smaller when entering the system as cold air, and larger as a heat carrier between the stack and the room. The resistance is thereby appreciably modified, and the sums of

pressure losses for runs of uniform temperature should be corrected by the factors given for this purpose. In most instances, however, the decreased volume and pressure loss for the cold air ducts are about offset by the increased loss in the heat flue, so that the final result without corrections, is likely to come within the limit of error from other causes. It is only in odd and extreme cases that it will pay to take these factors into account.

Individual heating or vent flues, each with separate air intake or outlet, may be figured without schedule. For apparatus with a duct system to be equalized for accurate distribution, the same mode of procedure is indicated as suggested for forced ventilating problems. The schedule of runs should, in addition, give the temperatures in each part of the system. Only the flues, as creating their own pressure, may be sized tentatively on the basis of the probable actual velocity resulting, which may differ radically according to the head available. The main ducts connecting them, on the other hand, should be proportioned for an even rate of pressure loss, as for blast work, and assuming as a basis the rate of loss maintaining in the connection at the far end of the line which should be the full size of the flue.

The pressure available for each heat flue should be used up on the individual runs, if possible by the stacks and vertical flues beyond, so that the system of main ducts can be figured as being under definite initial pressure. If the throats or branch pieces do not give sufficient opportunity to equalize the flow to the branches, the horizontal connections to the stacks may be varied in size. The air will then be delivered at each stack casing under the same pressure, thus placing the lower neutral zone where it belongs—at the mean level of heating.

Example of a Gravity Vent.—The ventilating arrangement illustrated by Fig. 35 is intended for exhausting toilets, locker rooms, etc., connected by an open stairway with a large gymnasium above. The building is exposed on all sides. The apparatus shown is intended only for the heating season, when open windows are objected to.

Individual vent flues from the several rooms of the lower story, the only one to be ventilated, were found to provide insufficient head since the neutral zone, under liberal leakage, and with the upper hall connecting by two stairways, is likely to be at the mean level of exposure. The large area of roofing and skylight place this level above the gallery floor. For individual flues from

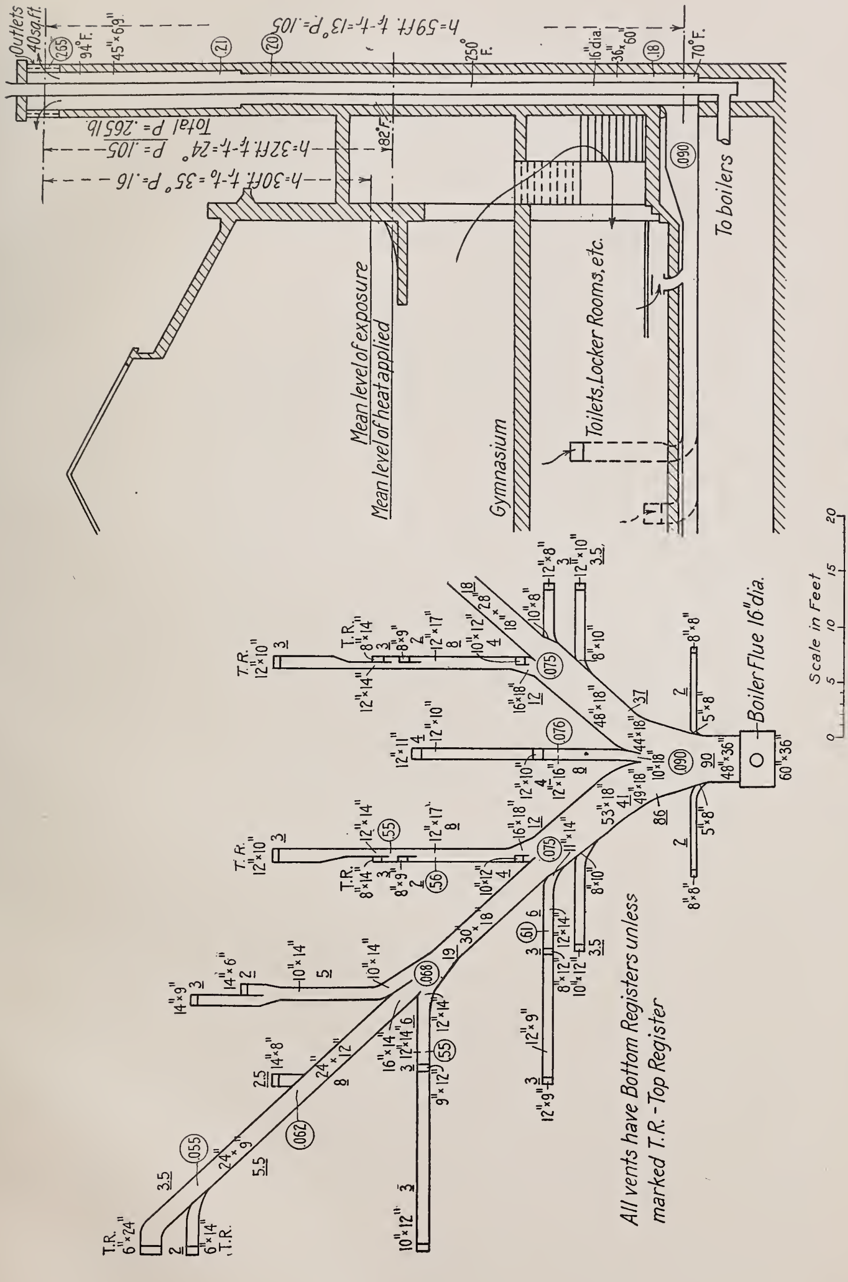


FIG. 35.—Examples of gravity ventilating system.

rooms located in the outside walls, and extending little beyond the main cornice, the effective head would be measured from the neutral zone to the outlets above the cornice. This small margin of effective height would be further reduced by cooling, which, at certain times, might offset it entirely and cause reversed action. A common shaft, which could be extended to a higher level promised better, but it became necessary to provide some extra draught power to overcome the resistance in the connecting ductwork. The smoke flue from the boiler was accordingly placed into this shaft, assuring an excess of temperature at least during the season when windows would be closed.

Without the heat of the boiler flue the draught power would be given by the temperature difference between the rooms and the outer air, in this case 35° F., and the height of the shaft from the neutral zone up to the outlet, 30 ft. above. The auxiliary chart gives .16 lb. per sq. ft. The additional buoyancy created by the boiler flue may be figured from the total temperature rise in the shaft, which is estimated to be 24° F. and the height of the shaft above the mean level at which the heat is applied, which is somewhat below the middle, or about 32 ft. The chart for heat flues, to be used in this case, gives .105 lb. additional pressure. The same result will be found by taking the entire height of shaft, 59 ft. and the mean temperature difference, which is slightly higher than half the total rise, corresponding to the greater heat emission in the lower half of the shaft. The total pressure available is therefore, $.16 + .105 = .265$ lb. per sq. ft., which must equal the sum of resistances in the ducts and the shaft, and give an approximate actual velocity of 7.5 ft. per sec. Allowing for resistance in the horizontal ducts and roughness of the brick flue the average speed was assumed to be somewhat smaller and the main duct and branches were proportioned for a rate of pressure loss of .04 lb. per sq. ft., per 100 ft., including corrections.

The calculation of the pressure losses, as shown on the schedule bore out these tentative assumptions very closely, the areas being corrected only for equalizing the suction. Allowances were made for higher temperature in the shaft, also for the increased ratio of $\frac{c}{a}$, the greater friction of brickwork and for shallow ducts.

Example of Indirect Heating System.—The schedule of such an apparatus represented by Fig. 36 gives as data for calculation, the

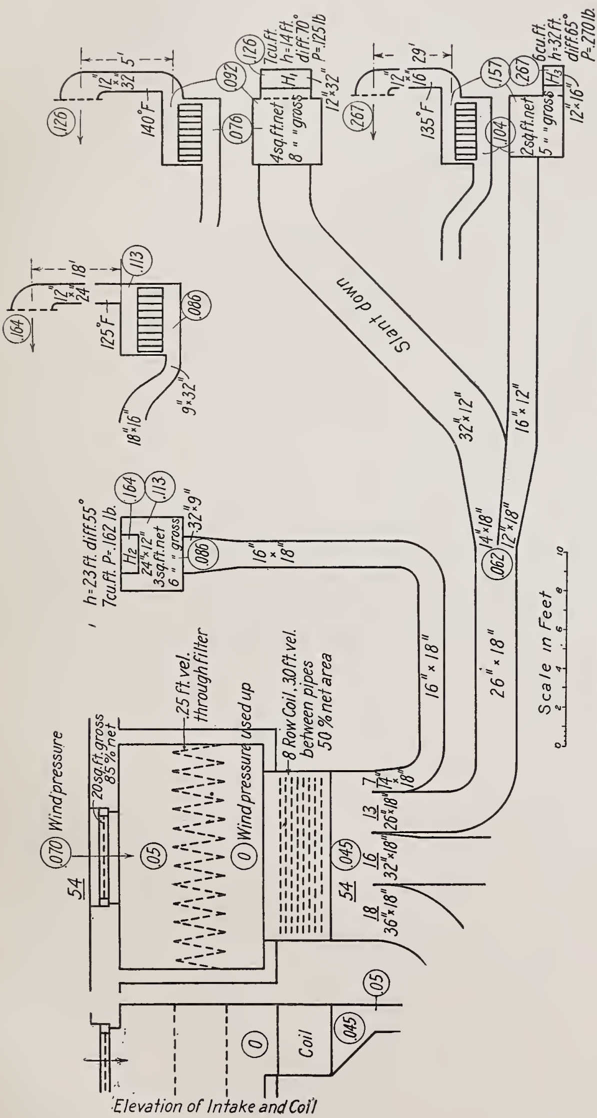


FIG. 36.—Example of duct system and flues for indirect heating.

volumes of air desired, the flue temperatures and the heights of draught for a group of indirect heating stacks connected to a common air intake. The windows of the respective rooms are facing the same points of compass as the intakes, so that the wind pressure can be depended upon at least to the extent of the allowance made for exposure for the room with the smallest head available. Assuming the latter to be 35 per cent., the heat required in still weather or the corresponding air volume is only 74 per cent. of the maximum amount, and the pressure to create the reduced flow figures only $.74^2 = .55$ of the total necessary to overcome the resistances of the filter, coil, ducts, stacks, and flues at full load. The balance of 45 per cent. would be made up, when needed, by wind pressure.

It is not safe, however, to figure on the aid of wind pressure to the full extent in every case, since the draught power is also affected by the flue temperature, which, in turn, is influenced by the speed of the air passing the stacks. Again, the heat supply to the surfaces may be reduced and thereby the buoyancy. It should be remembered further, that in a system with several flues having radically different available pressures, the distribution will be materially disturbed in the absence of wind, but sometimes this disadvantage must be accepted when the resistances are too great to be overcome by gravity alone, with reasonable flue sizes.

The effective pressures due to heat available for the three flues in the present example, as taken from the chart, are .125,.162, and .270 lb. respectively, the lowest head being naturally that for the first floor. In this case the .125 lb. represent about 64 per cent. of the total pressure that may be depended upon, which is .195 lb., including .07 lb. for wind pressure. This is counting on 25 per cent. for exposure, or a volume reduced to 80 per cent. in still weather. In order to assure fair distribution to the three rooms, the branch ducts leading to them are equalized as nearly as possible with the stated outside pressure and carrying the full volumes. The zone where the wind action is neutralized falls in this case between the filter and the tempering coil, allowing considerable leeway before an extra pressure would put the flues with greater resistance under a disadvantage. Of the varying total pressure for each flue an equal portion is used up to a certain point. The larger this portion, used in the common duct, the greater will be the relative difference of head left available for the

individual flues, and the greater their discrepancies in size. Without wind pressure the distribution is unbalanced the other way, the flues with the lightest draught being handicapped. It is possible, of course, to overcome these variations in delivery by temporary throttling, to suit weather conditions, or other forms of regulation, but a well balanced system will always require less attention in this respect and be more satisfactory. Pressures available, resistances, space, and other conditions should decide in each case how far the wind pressure ought to be depended upon.

If the various items of friction and obstruction in the example are checked up, it will be seen, that the entrance loss includes that of a screen on the end of a straight duct, plus the headway spent in entering the filter chamber. The entrance to the coil casing is a negligible item owing to the low velocity. The contraction to the size of the main duct, however, is taken into account. The inlets and outlets of stack casings have one side flush with the duct, and three sides at right angles, hence they will present only about two-thirds of the resistance of a flanged duct end. The stack sections are estimated to be equivalent to an 8-row coil with 50 per cent. free area, except that for the third floor where extra wide spacing is not needed. The latter is assumed equal to a 10-row coil with 50 per cent. clear space which is about equivalent to 8-rows with 40 per cent. net area.

The losses of head have been figured without allowance for temperature, since, in this case, the deduction would about offset the additions.